Energy Implications of Retrofitting Retail Sector Rooftop Units with Stepped-Speed and Variable-Speed Functionality

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Executive Summary

Background

According to the 2003 Commercial Buildings Energy Consumption Survey (EIA 2008), approximately 2.7 billion ft^2 of commercial retail floor space is served by packaged air-conditioning units. Often, these rooftop units (RTUs) are oversized to meet extreme and infrequent thermal loads (Felts et al. 2000). Thus, RTU supply fans may provide air at rates that are much higher than those needed to meet most thermal loads, wasting the energy needed to move and heat or cool the excess air.

To address this issue, some U.S. retailers have started to upgrade existing RTUs, retrofitting their constant-speed motors with stepped- or variable-speed functionality. Such retrofits may be more cost effective than replacing an entire RTU, particularly if the unit is in the middle of its lifespan. Other U.S. retailers, however, are uncomfortable pursuing this measure, as there is a lack of supporting data detailing the climate zone-specific energy savings potential associated with this upgrade. Building and portfolio energy managers have thus been unable to present a compelling business case for RTU fan motor upgrades.

This study uses whole-building energy simulation to estimate the energy impact of stepped- and variable-speed RTU fan motor retrofits in the retail environment, across 16 locations in all 15 U.S. climate zones. The results allow retailers to estimate the building-level energy savings associated with this retrofit measure. This is a critical step in enabling retailers to determine whether a compelling business case can be made.

Development Process

EnergyPlus Version 6.0 (DOE 2010) was used to evaluate the whole-building energy savings associated with stepped- and variable-speed RTU fan motor retrofits as follows:

1. Two prototype big-box retail EnergyPlus models were created: one with refrigeration systems and one without.
2. Each prototype model was replicated across 16 locations, which encompassed all 15 U.S. climate zones, as defined by DOE (2005).
3. Standard 90.1-2004 (ASHRAE 2004a) was applied to the building envelope to create climate-specific baseline models.
4. Custom sizing and control logic was used to modify a subset of the baseline models with stepped- and variable-speed RTU fan motor controls.
5. EnergyPlus was used to simulate the energy performance of all models to determine the energy savings associated with these retrofits.

Results

Table ES–1 through Table ES–3 present an overview of the simulation results[1]. Annual whole-building and electric energy savings were seen in all climate zones when switching to stepped- or variable-speed fan motor operation. Whole-building energy savings ranged from 0.7%–8.4%.

Normalized annual electricity savings ranged from 0.50–1.67 kWh/ft^2 (1.70–5.70 kBtu/ft^2). Average electricity savings across all locations were 1.17 kWh/ft^2 (3.98 kBtu/ft^2) for the stepped-speed cases and 1.23 kWh/ft^2 (4.40 kBtu/ft^2) for the variable-speed cases. The results suggest that for certain building types, annual whole-building electricity savings of 200,000 kWh may be realized.

Heating energy increased in all climates, because less fan energy was imparted to the supply airstream during the heating season. On average, this increase was 0.0149 therm/ft^2 for the stepped-speed cases and 0.0162 therm/ft^2 for the variable-speed cases. (This increase was always more than offset by the electricity use reduction.)

Based on the average energy savings, if variable-speed retrofits were implemented across 10% of the 2.7 billion ft^2 of retail space served by packaged RTUs (EIA 2008), retailers would save approximately 332 GWh/yr (1.13×10^{12} Btu/yr) of electricity and increase natural gas use by 4.4 million therm/yr (4.34×10^{11} Btu/yr). Assuming typical utility rates of $0.10/kWh and $1.00/therm, this would equate to annual utility cost savings of $28.8 million.

This measure requires that a relatively small number of RTU fan motors be upgraded per building. Therefore, this report makes a case that such retrofits should be investigated in all climate zones to address the oftentimes dispersed energy loads in commercial buildings.

The energy savings values presented here depend on a number of assumptions, including the cooling coil and fan flow rate sizing methodology, sequence of operations, space set points, and hours of operation. In order to provide broadly applicable findings, values were used that reflect common practice. Appendix C provides instructions about how to use the results of this study to estimate high-level utility cost savings. A retailer can use such results to justify site-specific assessments of retrofit feasibility and determine if a compelling business case can be made.

[1] Note that additional differences between the models with and without refrigeration, such as zoning layout and the fan flow rate sizing methodology, affect the energy consumption and savings values. Readers should exercise caution when comparing results between model types.

Table ES–1 Annual Whole-Building Energy Savings for All Models (%)

Climate Zone	Location	Models With Refrigeration		Models Without Refrigeration	
		Stepped-Speed	Variable-Speed	Stepped-Speed	Variable-Speed
1A	Miami, FL	2.5	3.5	2.3	2.3
2A	Houston, TX	3.2	4.0	3.6	3.6
2B	Phoenix, AZ	3.5	4.7	4.1	5.2
3A	Atlanta, GA	3.2	3.9	3.7	3.7
3B	Los Angeles, CA	6.1	6.5	8.1	8.3
3B	Las Vegas, NV	3.4	4.6	4.1	5.5
3C	San Francisco, CA	5.8	5.9	8.3	8.4
4A	Baltimore, MD	2.2	2.6	2.5	2.4
4B	Albuquerque, NM	3.1	3.8	3.6	4.1
4C	Seattle, WA	3.1	3.2	4.0	4.0
5A	Chicago, IL	1.8	2.1	2.1	2.0
5B	Boulder, CO	2.6	3.1	3.2	3.5
6A	Minneapolis, MN	1.5	1.8	1.7	1.7
6B	Helena, MT	1.8	2.1	2.3	2.4
7	Duluth, MN	1.1	1.2	1.4	1.3
8	Fairbanks, AK	0.7	0.7	0.9	0.9

Table ES–2 Annual Electricity Savings for All Models (kWh/ft^2)

Climate Zone	Location	Models With Refrigeration		Models Without Refrigeration	
		Stepped-Speed	Variable-Speed	Stepped-Speed	Variable-Speed
1A	Miami, FL	0.66	0.94	0.50	0.51
2A	Houston, TX	0.95	1.13	0.89	0.91
2B	Phoenix, AZ	1.00	1.30	0.98	1.22
3A	Atlanta, GA	1.14	1.30	1.11	1.13
3B	Los Angeles, CA	1.31	1.40	1.38	1.41
3B	Las Vegas, NV	1.11	1.39	1.13	1.41
3C	San Francisco, CA	1.47	1.50	1.66	1.67
4A	Baltimore, MD	1.19	1.31	1.16	1.24
4B	Albuquerque, NM	1.24	1.43	1.23	1.40
4C	Seattle, WA	1.41	1.44	1.55	1.57
5A	Chicago, IL	1.22	1.32	1.14	1.27
5B	Boulder, CO	1.30	1.42	1.29	1.44
6A	Minneapolis, MN	1.18	1.28	1.06	1.20
6B	Helena, MT	1.30	1.39	1.27	1.43
7	Duluth, MN	1.24	1.29	1.13	1.29
8	Fairbanks, AK	1.12	1.18	1.00	1.14

Table ES–3 Annual Natural Gas Savings* for All Models (therm/ft^2)

Climate Zone	Location	Models With Refrigeration		Models Without Refrigeration	
		Stepped-Speed	Variable-Speed	Stepped-Speed	Variable-Speed
1A	Miami, FL	0.000	0.000	0.000	0.000
2A	Houston, TX	–0.005	–0.005	–0.005	–0.005
2B	Phoenix, AZ	–0.004	–0.004	–0.005	–0.005
3A	Atlanta, GA	–0.012	–0.012	–0.012	–0.013
3B	Los Angeles, CA	–0.002	–0.002	–0.002	–0.002
3B	Las Vegas, NV	–0.009	–0.009	–0.010	–0.010
3C	San Francisco, CA	–0.009	–0.010	–0.011	–0.011
4A	Baltimore, MD	–0.020	–0.020	–0.020	–0.023
4B	Albuquerque, NM	–0.017	–0.017	–0.017	–0.019
4C	Seattle, WA	–0.023	–0.023	–0.025	–0.026
5A	Chicago, IL	–0.023	–0.024	–0.020	–0.025
5B	Boulder, CO	–0.021	–0.021	–0.019	–0.023
6A	Minneapolis, MN	–0.023	–0.023	–0.018	–0.023
6B	Helena, MT	–0.026	–0.026	–0.023	–0.027
7	Duluth, MN	–0.029	–0.029	–0.023	–0.029
8	Fairbanks, AK	–0.026	–0.028	–0.021	–0.025

*Negative values indicate that natural gas use increased.

Nomenclature

ACH	air changes per hour
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
c.i.	continuous insulation
cfm	cubic feet per minute
COP	coefficient of performance
DOE	U.S. Department of Energy
EPD	equipment power density
EUI	energy use intensity
ft	foot, feet
ft^2	square foot, square feet
HVAC	heating, ventilating, and air conditioning
in.	inch, inches
kBtu	kilo-Btu
kW	kilowatt, kilowatts
kWh	kilowatt-hour, kilowatt-hours
LPD	lighting power density
m	meter, meters
m^2	square meter, square meters
MJ	megajoule, megajoules
NREL	National Renewable Energy Laboratory
OA	outside air
RH	relative humidity
RTU	rooftop unit
SAT	supply air temperature
therm	unit of heat energy equal to 100 kBtu
VAV	variable air volume
w.c.	water column

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1.0 Introduction

According to the 2003 Commercial Buildings Energy Consumption Survey (EIA 2008), approximately 2.7 billion ft^2 of commercial retail floor space is served by packaged air-conditioning units. Often, these rooftop units (RTUs) are oversized to meet extreme and infrequent thermal loads (Felts et al. 2000). Thus, RTU supply fans may provide air at rates that are much higher than those needed to meet most thermal loads, wasting the energy needed to move and heat or cool the excess air.

To enable more efficient RTU operation during normal operating conditions, some U.S. retailers have started to upgrade existing RTUs, retrofitting their constant-speed motors with stepped- or variable-speed functionality. Such retrofits may be more cost effective than replacing an entire RTU, particularly if the unit is in the middle of its lifespan. Other U.S. retailers, however, are uncomfortable pursuing this measure, as there is a lack of supporting data detailing the climate zone-specific energy savings potential associated with this upgrade. Building and portfolio energy managers have thus been unable to present a compelling business case for RTU fan motor upgrades.

This study uses whole-building energy simulation to estimate the energy impact of stepped- and variable-speed RTU fan motor retrofits in the retail environment, across 16 locations in all 15 U.S. climate zones. The results allow retailers to estimate the building-level energy savings associated with this retrofit measure. This is a critical step in enabling retailers to determine whether a compelling business case can be made.

1.1 Objective

Our objective was to use whole-building energy simulation to quantify the energy savings associated with stepped- and variable-speed RTU fan motor retrofits in a typical big-box retail environment.

1.2 Scope

This report provides normalized, simulation-based energy savings estimates for RTU fan motor retrofits. Because the big-box retail environment is diverse, we examined buildings with and without refrigeration systems (walk-in coolers and refrigerated cases).

2.0 Building Energy Modeling Methodology

2.1 EnergyPlus

EnergyPlus Version 6.0 (DOE 2010), a publicly available building energy simulation engine, was used for all analyses. EnergyPlus was selected because it is a detailed U.S. Department of Energy (DOE) simulation tool that computes building energy use based on the interactions between climate; building form and fabric; internal gains; heating, ventilating, and air conditioning (HVAC) systems; and renewable energy systems. All simulations were run on desktop computers at the National Renewable Energy Laboratory (NREL).

2.2 Climate Zones

The models used for this study were simulated in 16 locations, which represent all 15 U.S. climate zones. The eight climate zones and three subzones used to determine these locations are depicted in Figure 2–1. The zones are defined primarily by heating degree days and cooling degree days (Briggs et al. 2003). The climate zones range from hot (zone 1) to cold (zone 8). Subzones indicate varying moisture conditions. Humid subzones are designated by the letter A, dry subzones by B, and marine subzones by C.

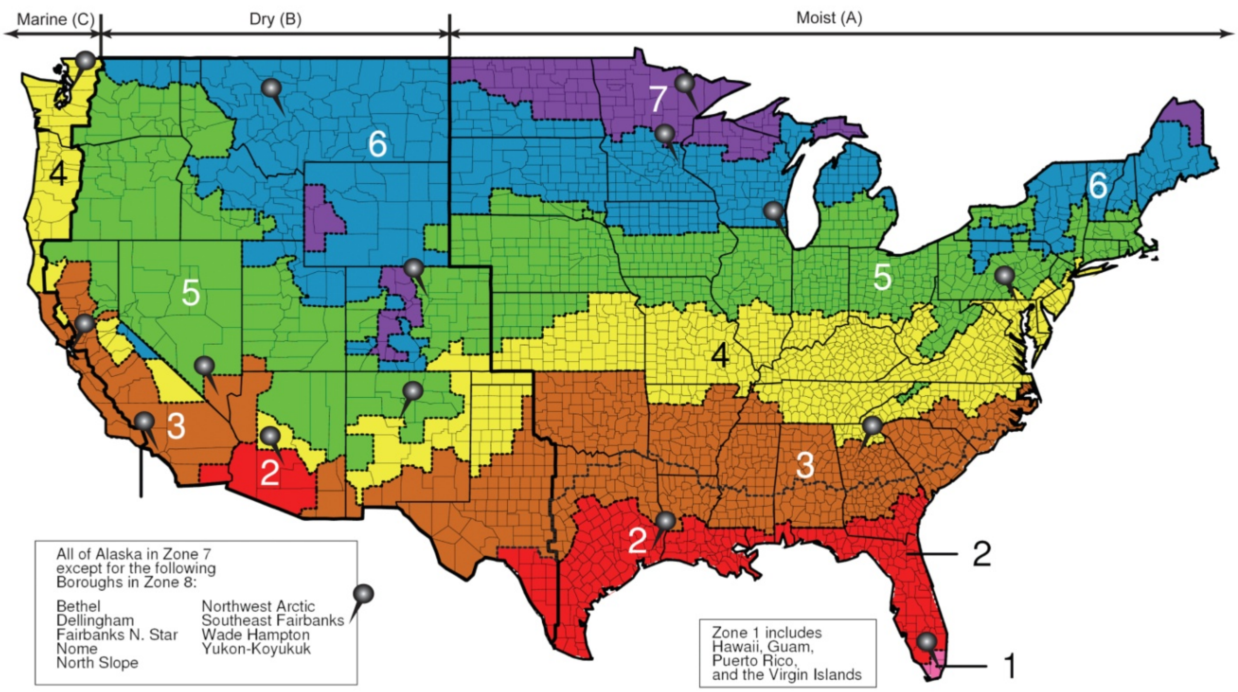

Figure 2–1 DOE climate zones and representative cities
(Credit: DOE 2005)

Models were simulated for large cities only, as the weather data for such locations were directly applicable to a large fraction of the total U.S. building floor area. Energy savings were determined by running the baseline and all retrofit simulations with the same Typical

Meteorological Year 3 weather file (one set of simulations for each city) (Deru et al. 2011). The 16 specific locations for which analyses were performed are listed below, are marked in Figure 2–1, and are designated as being representative of their respective climate zones.[2]

- Zone 1A Miami, Florida (hot, humid)
- Zone 2A Houston, Texas (hot, humid)
- Zone 2B Phoenix, Arizona (hot, dry)
- Zone 3A Atlanta, Georgia (hot, humid)
- Zone 3B Las Vegas, Nevada (hot, dry) and Los Angeles, California (warm, dry)
- Zone 3C San Francisco, California (marine)
- Zone 4A Baltimore, Maryland (mild, humid)
- Zone 4B Albuquerque, New Mexico (mild, dry)
- Zone 4C Seattle, Washington (marine)
- Zone 5A Chicago, Illinois (cold, humid)
- Zone 5B Boulder, Colorado (cold, dry)
- Zone 6A Minneapolis, Minnesota (cold, humid)
- Zone 6B Helena, Montana (cold, dry)
- Zone 7 Duluth, Minnesota (very cold)
- Zone 8 Fairbanks, Alaska (extremely cold)

2.3 Modeling Process

Two prototype big-box retail EnergyPlus models were created: one with refrigeration systems (for food sales) and one without. Each prototype model was then replicated across 16 locations, and Standard 90.1-2004 (ASHRAE 2004a) was applied to the building envelope to create climate-specific baseline models. Each of the 32 baseline models was then modified with stepped- and variable-speed RTU fan motor functionality, and the energy performance of all 96 models (32 constant-speed, 32 stepped-speed, and 32 variable-speed) was simulated using EnergyPlus to determine energy savings.

2 This report contains numerous climate zone tables in which the states will be designated with their respective two-letter postal codes: Florida (FL), Texas (TX), Arizona (AZ), Georgia (GA), Nevada (NV), California (CA), Maryland (MD), New Mexico (NM), Washington (WA), Illinois (IL), Colorado (CO), Minnesota (MN), Montana (MT), and Alaska (AK).

3.0 Model Development

3.1 Form

The building form for each prototype model was based on floor plans supplied by a major U.S. retailer. The models with refrigeration included all space types in the models without refrigeration, plus a pharmacy, a grocery area, a walk-in freezer area, and a kitchen. The total floor area for the models with refrigeration was 133,275 ft^2 (12,382 m^2). The total floor area for the models without refrigeration was 111,825 ft^2 (10,389 m^2). Figure 3–1 through Figure 3–4 and Table 3–1 and Table 3–2 provide additional details about the zone layout and space types assumed for both prototype models.

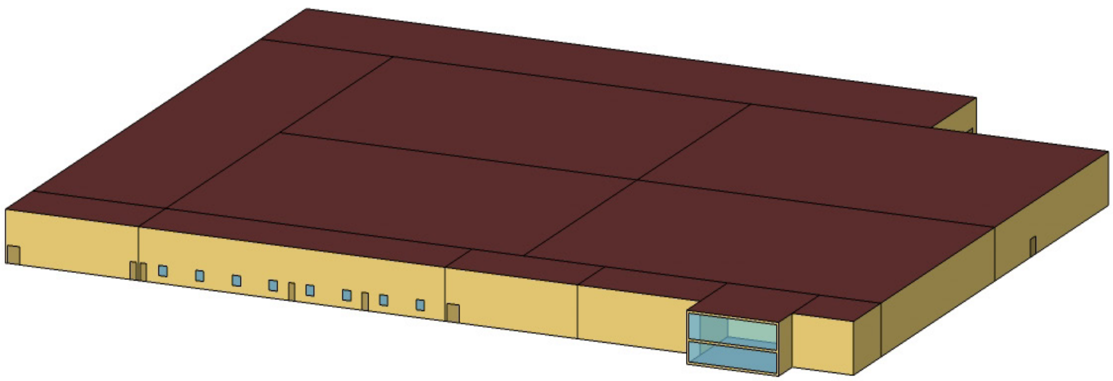

Figure 3–1 Rendering of models with refrigeration: view from the southeast
(Credit: Rachel Romero/NREL)

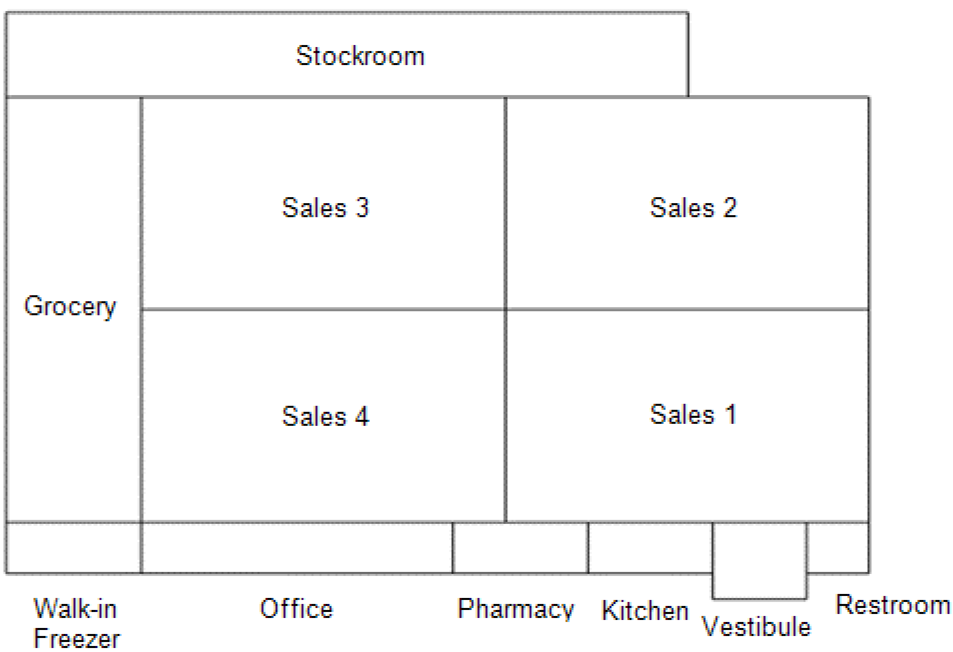

Figure 3–2 Floor plan for models with refrigeration
(Credit: Rachel Romero/NREL)

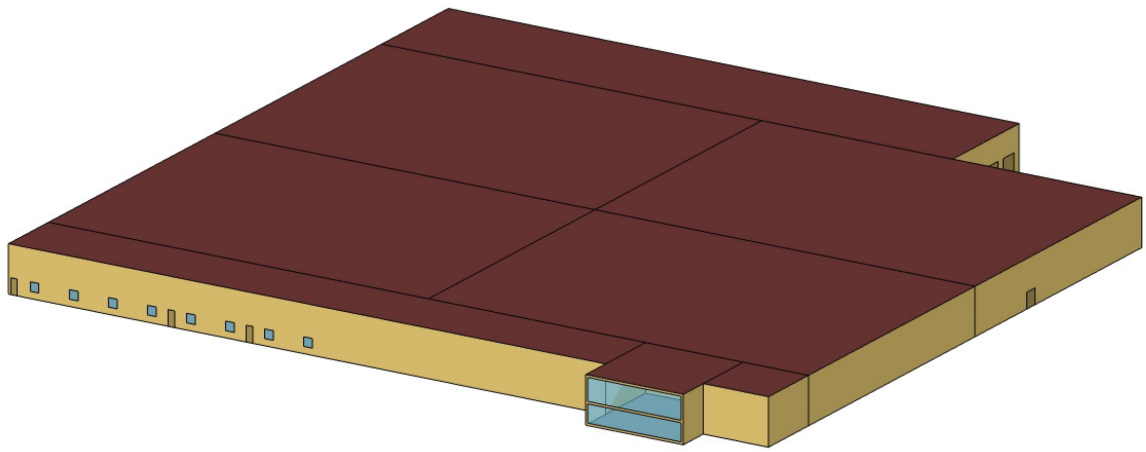

Figure 3–3 Rendering of models without refrigeration: view from the southeast
(Credit: Rachel Romero/NREL)

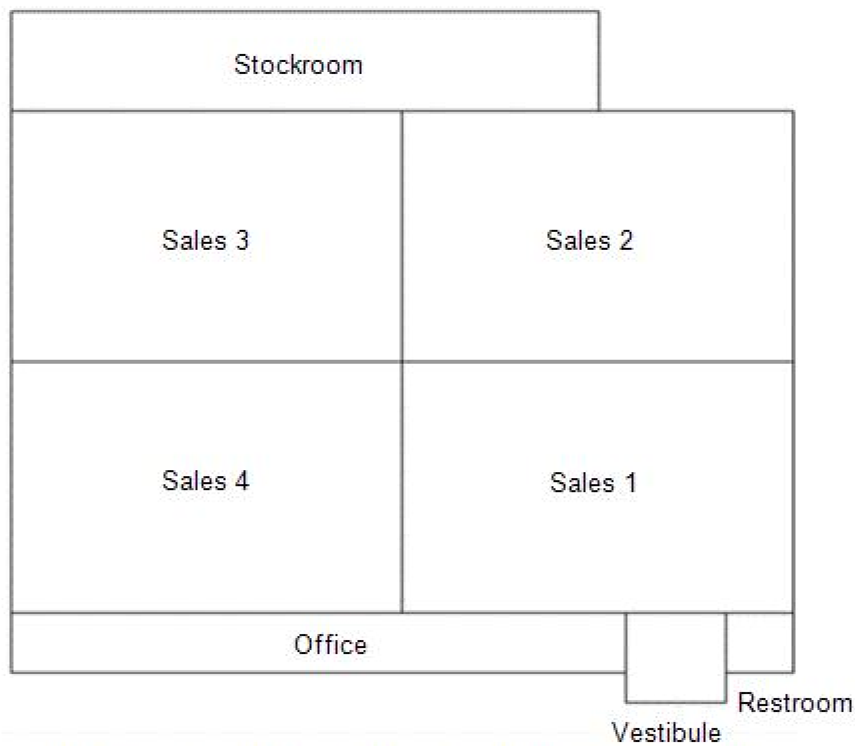

Figure 3–4 Floor plan for models without refrigeration
(Credit: Rachel Romero/NREL)

Table 3–1 Space Type Details for Models With Refrigeration

Zone	Floor Area (ft^2)	Floor Area (m^2)	Total (%)
Grocery	16,250	1,510	12.2
Kitchen	1,800	167	1.4
Office	4,500	418	3.4
Pharmacy	1,950	181	1.5
Restroom	900	84	0.7
Sales 1	21,875	2,032	16.4
Sales 2	21,875	2,032	16.4
Sales 3	21,875	2,032	16.4
Sales 4	21,875	2,032	16.4
Stockroom	16,400	1,524	12.3
Vestibule	2,025	188	1.5
Walk-in freezer	1,950	181	1.5
Total	**133,275**	**12,382**	**100.0**

Table 3–2 Space Type Details for Models Without Refrigeration

Zone	Floor Area (ft^2)	Floor Area (m^2)	Total (%)
Office	8,250	766	7.4
Restroom	900	84	0.8
Sales 1	21,875	2,032	19.6
Sales 2	21,875	2,032	19.6
Sales 3	21,875	2,032	19.6
Sales 4	21,875	2,032	19.6
Stockroom	13,150	1,222	11.8
Vestibule	2,025	188	1.8
Total	**111,825**	**10,389**	**100.0**

3.2 Fabric

3.2.1 Opaque Envelope

The thermal properties for opaque exterior surfaces in all models were taken from Standard 90.1-2004 (ASHRAE 2004a). Wall insulation values were chosen assuming steel-framed construction. The roof insulation was assumed to be entirely above deck. Table 3–3 details the thermal resistance of the exterior walls and roofs for all models by climate zone.

Table 3–3 Envelope Details

Climate Zone	Location	Exterior Wall Insulation ($ft^2·h·°F/Btu$)	Roof Insulation ($ft^2·h·°F/Btu$)
1A	Miami, FL	R-13	R-15 c.i.
2A	Houston, TX	R-13	R-15 c.i.
2B	Phoenix, AZ	R-13	R-15 c.i.
3A	Atlanta, GA	R-13	R-15 c.i.
3B	Los Angeles, CA	R-13	R-15 c.i.
3B	Las Vegas, NV	R-13	R-15 c.i.
3C	San Francisco, CA	R-13	R-15 c.i.
4A	Baltimore, MD	R-13	R-15 c.i.
4B	Albuquerque, NM	R-13	R-15 c.i.
4C	Seattle, WA	R-13	R-15 c.i.
5A	Chicago, IL	R-13 + R-3.8 c.i.	R-15 c.i.
5B	Boulder, CO	R-13 + R-3.8 c.i.	R-15 c.i.
6A	Minneapolis, MN	R-13 + R-3.8 c.i.	R-15 c.i.
6B	Helena, MT	R-13 + R-3.8 c.i.	R-15 c.i.
7	Duluth, MN	R-13 + R-7.5 c.i.	R-15 c.i.
8	Fairbanks, AK	R-13 + R-7.5 c.i.	R-20 c.i.

3.2.2 Vertical Fenestration

Window properties for all models were determined according to Standard 90.1-2004 (ASHRAE 2004a) and are detailed in Table 3–4.

Table 3–4 Fenestration Details

Climate Zone	U-Factor ($Btu/h·ft^2·°F$)	Solar Heat Gain Coefficient	Visible Light Transmittance
1A	U-1.22	0.25	0.25
2 (A,B)	U-1.22	0.25	0.25
3 (A,B)	U-0.57	0.39	0.50
3C	U-1.22	0.61	0.61
4 (A,B,C)	U-0.57	0.39	0.50
5 (A,B)	U-0.57	0.49	0.62
6 (A,B)	U-0.57	0.49	0.62
7	U-0.57	0.49	0.49
8	U-0.46	0.45	0.45

3.3 Infiltration

The infiltration rate for all zones in all models was assumed to be 0.15 cfm/ft^2 at 0.016 in. w.c. (0.76 $L/s·m^2$ at 4 Pa) of above-grade envelope area (walls and roof). This value was chosen as it represented an acceptable middle ground between the infiltration rates used by Deru et al. (2011) for new construction and existing buildings. During occupied hours, when outdoor air (OA) was being brought in by the HVAC system, the building was assumed to be pressurized, and the infiltration rate was set to 50% of this design value. During unoccupied hours, the infiltration rate was reset to the design value.

Additional infiltration was added to the vestibule zone at a maximum flow rate of 2.22 cfm/ft^2 (11.30 L/s·m^2) of floor area to account for infiltration associated with opening and closing of the main entry doors. This value was calculated using Equation 52 from Chapter 16 of the *2009 ASHRAE Handbook—Fundamentals* (ASHRAE 2009):

$$Q = C_A \times A \times R_p \tag{3-1}$$

Where:

Q	=	airflow rate (cfm)
C_A	=	airflow coefficient [cfm/ft^2/(in. of water)$^{0.5}$]
A	=	area of the door opening (ft^2)
R_p	=	pressure factor [(in. of water)$^{0.5}$]

The airflow coefficient and pressure factor were calculated using methodology presented by Yuill (1996). To determine the airflow coefficient, a maximum occupancy flow rate through the entry of 159 people per hour was assumed, a value based on measured data presented by Yuill (1996). The door opening was assumed to be equal to 42 ft^2 (3.9 m^2). Daily variations in vestibule infiltration were assumed to follow the occupancy schedule.

3.4 Internal Load Densities

3.4.1 Occupancy

Peak occupant density values were taken from Standard 62.1-2004 (ASHRAE 2004b). For the models with refrigeration, the average floor area per person was 76.5 ft^2 (7.1 m^2); for those without refrigeration, it was 74.6 ft^2 (6.9 m^2). Table 3–5 and Table 3–6 detail the peak occupant density by zone for both model types.

Table 3–5 Peak Occupant Density for Models With Refrigeration

Zone	Standard 62.1-2004 Equivalent Zone	Occupancy Density (people/100 m^2)	Area per Person (ft^2/person)	Area per Person (m^2/person)
Grocery	Supermarket	8.0	134.5	12.5
Kitchen	Sales	15.0	71.8	6.7
Office	Office	5.0	215.3	20.0
Pharmacy	Pharmacy	10.0	107.6	10.0
Restroom	Sales	15.0	71.8	6.7
Sales 1	Sales	15.0	71.8	6.7
Sales 2	Sales	15.0	71.8	6.7
Sales 3	Sales	15.0	71.8	6.7
Sales 4	Sales	15.0	71.8	6.7
Stockroom	Shipping/receiving	0.0	0.0	0.0
Vestibule	Main entry lobbies	10.0	107.6	10.0
Walk-in freezer	Sales	15.0	71.8	6.7
Total		**11.8**	**76.5**	**7.1**

Table 3–6 Peak Occupant Density for Models Without Refrigeration

Zone	Standard 62.1-2004 Equivalent Zone	Occupancy Density (people/100 m^2)	Area per Person (ft^2/person)	Area per Person (m^2/person)
Office	Office	5.0	215.3	20.0
Restroom	Sales	15.0	71.8	6.7
Sales 1	Sales	15.0	71.8	6.7
Sales 2	Sales	15.0	71.8	6.7
Sales 3	Sales	15.0	71.8	6.7
Sales 4	Sales	15.0	71.8	6.7
Stockroom	Shipping/receiving	0.0	0.0	0.0
Vestibule	Main entry lobbies	10.0	107.6	10.0
Total		**12.4**	**74.6**	**6.9**

3.4.2 Plug and Process Loads

The equipment data detailed in Table 3–7 and Table 3–8 were adapted from metered values obtained from a big-box retailer. The total equipment power density (EPD) for the models with refrigeration was 0.65 W/ft^2 (7.0 W/m^2); those without refrigeration had an EPD of 0.56 W/ft^2 (6.0 W/m^2). This difference is due mainly to the lack of a "kitchen" zone in the models without refrigeration.

Table 3–7 EPDs for Models With Refrigeration

Zone	EPD (W/ft^2)	EPD (W/m^2)
Grocery	0.00	0.00
Kitchen	14.08	151.59
Office	1.00	10.72
Pharmacy	0.35	3.75
Restroom	0.00	0.00
Sales 1	0.55	5.91
Sales 2	0.55	5.91
Sales 3	0.55	5.91
Sales 4	0.55	5.91
Stockroom	0.50	5.34
Vestibule	0.00	0.00
Walk-in freezer	0.00	0.00
Total	**0.65**	**7.00**

Table 3–8 EPDs for Models Without Refrigeration

Zone	EPD (W/ft^2)	EPD (W/m^2)
Office	1.00	10.72
Restroom	0.00	0.00
Sales 1	0.55	5.91
Sales 2	0.55	5.91
Sales 3	0.55	5.91
Sales 4	0.55	5.91
Stockroom	0.50	5.34
Vestibule	0.00	0.00
Total	**0.56**	**6.04**

3.4.3 Refrigeration Loads

For the models with refrigeration, refrigerated cases were assumed to be present in the grocery, kitchen, and walk-in freezer zones. The grocery zone contained single-deck ice cream cases, vertical door freezer cases for frozen foods, and multideck cases for dairy and deli items, for a total of 415 linear feet (126.5 m) of refrigerated cases. The kitchen zone contained 20 linear feet (6.1 m) of multideck cases for dairy and deli items. The walk-in freezer zone contained 1,779 ft^2 (165.3 m^2) of walk-in freezers.

3.5 Schedules

All models assumed that the buildings were open to customers from 8:00 a.m. to 10:00 p.m., seven days per week; however, because store employees had to be in the space before and after sales hours, occupied hours were assumed to be from 7:00 a.m. to 11:00 p.m., seven days per week. Occupancy, lighting, equipment, and set point schedules are presented in Appendix A.

3.6 Lighting

Lighting power density (LPD) values for all models were taken from Standard 90.1-2004 Table 9.6.1 (ASHRAE 2004a). The total building LPD for the models with refrigeration was 1.53 W/ft^2 (16.5 W/m^2). The building LPD for the models without refrigeration was 1.52 W/ft^2 (16.4 W/m^2). The LPDs for each model type and zone and the associated Standard 90.1-2004 (ASHRAE 2004a) space type are listed in Table 3–9 and Table 3–10.

All models were assumed to have 3,014 W of exterior lighting. This value was adapted from data obtained from a big-box retailer. The exterior lighting system was controlled by an ambient light sensor, ensuring that the lights were on from dusk until dawn. Exterior lighting was modeled to more accurately determine building peak electrical demand. It had no impact on the results of this study.

Table 3–9 LPDs for Models With Refrigeration

Zone	Standard 90.1-2004 Space Type	LPD (W/ft^2)	LPD (W/m^2)
Grocery	Retail-sales area	1.7	18.3
Kitchen	Food preparation	1.2	12.9
Office	Office	1.1	11.8
Pharmacy	Hospital-pharmacy	1.2	12.9
Restroom	Restrooms	0.9	9.7
Sales 1	Retail-sales area	1.7	18.3
Sales 2	Retail-sales area	1.7	18.3
Sales 3	Retail-sales area	1.7	18.3
Sales 4	Retail-sales area	1.7	18.3
Stockroom	Active storage	0.8	8.6
Vestibule	Atrium	0.6	6.5
Walk-in freezer	Retail-sales area	1.7	18.3
Total		**1.53**	**16.5**

Table 3–10 LPDs for Models Without Refrigeration

Zone	Standard 90.1-2004 Space Type	LPD (W/ft^2)	LPD (W/m^2)
Office	Office	1.1	11.8
Restroom	Restrooms	0.9	9.7
Sales 1	Retail-sales area	1.7	18.3
Sales 2	Retail-sales area	1.7	18.3
Sales 3	Retail-sales area	1.7	18.3
Sales 4	Retail-sales area	1.7	18.3
Stockroom	Active storage	0.8	8.6
Vestibule	Atrium	0.6	6.5
Total		**1.52**	**16.4**

3.7 Heating, Ventilating, and Air Conditioning

3.7.1 System Type

Each zone was assumed to be conditioned by a packaged RTU. Each RTU model consisted of a fan, a two-stage direct expansion cooling coil, a single-stage gas-fired heat exchanger, and an OA mixer.

3.7.2 Rooftop Unit Performance Properties

Performance properties for all RTUs are shown in Table 3–11. The energy efficiency ratio was taken from Standard 90.1-2004 Table 6.8.1A (ASHRAE 2004a). The heating efficiency, heating fuel type, and economizer control type were taken from previous RTU-related work (Hale et al. 2009). Additional assumptions included an internal static pressure drop of 0.42 in. w.c. (105 Pa), an external static pressure drop of 0.70 in. w.c. (174 Pa), and a total fan efficiency of 35%. These assumptions were based on previous engineering experience. Combined coefficients of performance (COPs) for the compressor and condenser were calculated for the models with refrigeration and the models without refrigeration based on their respective design fan flow rates (see Section 3.7.4) and the above values for energy efficiency ratio, fan efficiency, and fan static pressure drop. Because of the differences in design fan flow rate (see Section 3.7.4) between the models with and without refrigeration, separate COP values for each model type were calculated and used to ensure that the overall RTU energy efficiency ratio met that specified by Standard 90.1-2004 Table 6.8.1A (ASHRAE 2004a). For further information, see the methodology detailed in Hale et al.

Table 3–11 HVAC System Performance Values

HVAC Input	Modeled Value
RTU energy efficiency ratio	10.1
Compressor/condenser combined COP for models with refrigeration	3.45
Compressor/condenser combined COP for models without refrigeration	3.36
Heating efficiency	80%
Heating fuel	Natural gas
Fan static pressure	1.12 in. w.c. (279 Pa)
Fan mechanical efficiency: power gained by the air/power input to the fan motor	35%
Combined fan motor and belt efficiency	77%
Economizer control	Differential dry-bulb

Five empirically derived curve fits were used to model the performance map of the direct expansion equipment. These described the unit's cooling capacity and efficiency as functions of the cooling coil inlet air conditions, cooling coil airflow rate, and the condenser inlet air dry-bulb temperature. The specific performance curves used for this study were derived from measured values obtained through laboratory testing of a 10-ton RTU (Kozubal et al. 2010).

3.7.3 Economizer

Differential dry-bulb economizers were included in all models and across all climate zones. The OA damper was modeled as having full modulating capability.

3.7.4 Sizing

EnergyPlus Runtime Language was used to size the cooling coil capacity and fan flow rate for each RTU. Cooling coil capacities for all models were sized to provide one ton of cooling for every 375 ft^2 served (100.9 W/m^2). Fan sizes, in terms of maximum volumetric flow rate, were set to 1.2 cfm/ft^2 (6.1 $L/s{\cdot}m^2$) of zone floor area for the models with refrigeration and to 1.0 cfm/ft^2 (5.1 $L/s{\cdot}m^2$) of zone floor area for the models without refrigeration. These values were based on engineering judgment and are meant to represent the typical RTU fan over sizing observed in the retail sector. The heating coils were "autosized" using EnergyPlus' built-in sizing algorithms.

We assumed that only 10-ton RTUs were used to condition the building. For example, if a zone required 20 tons of cooling, a single 20-ton RTU was simulated using 10-ton unit performance properties, thereby mimicking the performance of two, 10-ton units operating together to serve that zone.

3.7.5 Outdoor Air

OA supply flow rates for each zone were calculated in accordance with Standard 62.1-2004 (ASHRAE 2004b). Table 3–12 and Table 3–13 detail the OA design flow rates for both prototype models.

Table 3–12 OA Rates for Models With Refrigeration

Zone	Standard 62.1-2004 Equivalent Zone	OA per Area (cfm/ft^2)	OA per Area ($L/s{\cdot}m^2$)	OA per Person (cfm/person)	OA per Person (L/s·person)	Total OA (cfm)	Total OA (L/s)
Grocery	Supermarket	0.06	0.305	7.5	3.540	1,881	888
Kitchen	Sales	0.12	0.610	7.5	3.540	404	191
Office	Office	0.06	0.305	5.0	2.360	375	177
Pharmacy	Pharmacy	0.18	0.914	5.0	2.360	442	208
Restroom	Sales	0.12	0.610	7.5	3.540	202	95
Sales 1	Sales	0.12	0.610	7.5	3.540	4,911	2,318
Sales 2	Sales	0.12	0.610	7.5	3.540	4,911	2,318
Sales 3	Sales	0.12	0.610	7.5	3.540	4,911	2,318
Sales 4	Sales	0.12	0.610	7.5	3.540	4,911	2,318
Stockroom	Shipping/receiving	0.12	0.610	0.0	0.000	1,968	929
Vestibule	Main entry lobbies	0.06	0.305	5.0	2.360	216	102
Walk-in freezer	Sales	0.12	0.610	7.5	3.540	438	207
Total		**0.11**	**0.562**	**6.4**	**3.029**	**25,569**	**12,068**

Table 3–13 OA Rates for Models Without Refrigeration

Zone	Standard 62.1-2004 Equivalent Zone	OA per Area (cfm/ft^2)	OA per Area ($L/s{\cdot}m^2$)	OA per Person (cfm/person)	OA per Person (L/s·person)	Total OA (cfm)	Total OA (L/s)
Office	Office	0.06	0.305	5.0	2.360	375	177
Restroom	Sales	0.12	0.610	7.5	3.540	202	95
Sales 1	Sales	0.12	0.610	7.5	3.540	4,911	2,318
Sales 2	Sales	0.12	0.610	7.5	3.540	4,911	2,318
Sales 3	Sales	0.12	0.610	7.5	3.540	4,911	2,318
Sales 4	Sales	0.12	0.610	7.5	3.540	4,911	2,318
Stockroom	Shipping/receiving	0.12	0.610	0.0	0.000	1,968	929
Vestibule	Main entry lobbies	0.06	0.305	5.0	2.360	216	102
Total		**0.11**	**0.582**	**6.4**	**3.015**	**22,405**	**10,574**

3.7.6 Supply Fan

Each RTU supply fan was hard-sized to provide a maximum volumetric flow rate of 1.2 cfm/ft^2 (6.1 $L/s{\cdot}m^2$) of zone floor area for the models with refrigeration and 1.0 cfm/ft^2 (5.1 $L/s{\cdot}m^2$) of zone floor area for the models without refrigeration. The fan power at part-load was modeled according to ASHRAE 90.1-2004 Appendix G Table G3.1.3.15 Method 2 for the Part-Load Fan Power Equation (ASHRAE 2004a):

$$P_{fan} = 0.0013 + 0.1470 \times PLR_{fan} + 0.9506 \times (PLR_{fan})^2 - 0.0998 \times (PLR_{fan})^3 \quad (3\text{-}2)$$

Where:

P_{fan} = fraction of full-load fan power

PLR_{fan} = fan part-load ratio (current cfm/design cfm)

For all models with refrigeration, the RTU fan minimum flow rate was 0.7 cfm/ft^2 (3.7 L/s·m^2) of zone floor area, a value equivalent to 60% of the design flow rate. The minimum RTU fan flow rate for the models without refrigeration was 0.4 cfm/ft^2 (2.0 L/s·m^2) of zone floor area, a value equivalent to 40% of the design flow rate. The RTU fan minimum flow rate was kept higher in the models with refrigeration to prevent microclimates that the refrigerated cases would otherwise have created.

3.7.7 Control Logic

Typically, EnergyPlus models are run with simulation time steps set between 10 and 15 minutes, which is longer than the time the HVAC system normally takes to respond to changing building conditions. At the beginning of each time step, EnergyPlus' traditional controls predict the exact amount of heating or cooling that will be needed for each thermal zone to reach the space set point. EnergyPlus then compares this value to the HVAC system capacity and runs the system at an artificial heating or cooling part-load ratio—taking into account coil cycling losses—to ensure that at the end of the time step, the zone temperature meets the space set point.

The downside to this type of control is that the fan is locked into a single, average operational flow rate during each 10- to 15-minute time step. Although this control scheme will accurately predict annual fan energy use, it cannot be used to compare the effects of certain dynamic fan operation strategies, as EnergyPlus limits the types of available fan control schemes.

To resolve this issue, custom EnergyPlus Runtime Language code was written to control RTU fan motor and coil operation, which enabled the study of stepped- and variable-speed RTU operation. The simulation time steps used for this study were set equal to 1 minute. Smaller time steps allowed us to more realistically reflect real-world stepped- and variable-speed RTU operation, where fan flow rates are adjusted on minute timescales in response to changing indoor, outdoor, and coil conditions.

The following subsections describe the EnergyPlus Runtime Language code that was used to control each RTU fan motor type. This code was adapted from real-world control logic to characterize the energy performance of the investigated strategies across a broad array of building types and climates. As such, the control logic provided here was not tailored to a particular application; changes to the control logic for a particular installation may provide even greater energy savings than those presented in this report.

3.7.7.1 Constant-Speed Case

For the baseline case (constant-speed), the RTU fan speed was fixed at the design fan speed during occupied hours for all operating modes. The control logic narrative for the constant-speed fan motor case follows.

1. Deadband
    a. During occupied hours, when the system was not in heating, economizing, or cooling mode, deadband mode was active.
        i. The OA dampers were set to the minimum ventilation position.
        ii. The heating and cooling coils were turned off.

   b. During unoccupied hours, when the system was not in heating, economizing, or cooling mode, the RTU fan was turned off.
2. Heating
   a. The heating coil was engaged when the zone temperature was below the heating set point ($T_{zone} < T_{set,heat}$).
      i. During occupied hours, the OA dampers were set to the minimum ventilation position.
      ii. During unoccupied hours, the OA dampers were closed.
      iii. The RTU continued to heat the zone until the zone temperature was at least 1°F above the heating set point [$T_{zone} \geq (T_{set,heat} + 1°F)$].
3. Economizing
   a. If (1) the OA temperature was less than 65°F and greater than 50°F ($50°F < T_{OA} < 65°F$), and (2) the zone temperature was above the cooling set point by no more than 1°F [$T_{set,cool} < T_{zone} \leq (T_{set,cool} + 1°F)$], economizing was engaged.
      i. The OA dampers were set to 100% open.
      ii. The cooling coils remained off.
      iii. The RTU operated in economizer mode until either:
         1. The OA temperature was less than or equal to 50°F ($T_{OA} \leq 50°F$), at which point the RTU entered Stage 1 cooling mode.
         2. The OA temperature was greater than or equal to 65°F ($T_{OA} \geq 65°F$), at which point the RTU entered Stage 1 cooling mode.
         3. The zone temperature was at least 1°F below the cooling set point [$T_{zone} \leq (T_{set,cool} - 1°F)$], at which point the RTU entered deadband mode.
         4. The zone temperature was greater than 1°F above the cooling set point [$T_{zone} > (T_{set,cool} + 1°F)$], at which point the RTU entered Stage 2 cooling mode.
4. Cooling
   a. Stage 1 cooling was engaged when the zone temperature rose above the cooling set point by no more than 1°F [$T_{set,cool} < T_{zone} \leq (T_{set,cool} + 1°F)$] and the OA temperature was not between 50°F and 65°F (economizing condition).
      i. The first compressor was engaged.
      ii. The RTU remained in Stage 1 cooling until either:
         1. The zone temperature was at least 1°F below the cooling set point [$T_{zone} \leq (T_{set,cool} - 1°F)$], at which point the RTU entered deadband mode.
         2. The zone temperature was greater than 1°F above the cooling set point [$T_{zone} > (T_{set,cool} + 1°F)$], at which point the RTU entered Stage 2 cooling mode.
      iii. During unoccupied hours, the OA dampers were closed.
   b. Stage 2 cooling was engaged when the zone temperature was greater than 1°F above the cooling set point [$T_{zone} > (T_{set,cool} + 1°F)$].
      i. Both compressors were engaged.
      ii. The RTU remained in Stage 2 cooling until the zone temperature was at least 1°F below the cooling set point [$T_{zone} \leq (T_{set,cool} - 1°F)$], at which point the RTU entered deadband mode.

    iii. During unoccupied hours, the OA dampers were closed, unless the OA temperature was less than 65°F and greater than 50°F (50°F < T_{OA} < 65°F), in which case the OA dampers were set to the minimum ventilation position.
  c. During occupied hours, the OA dampers were set to the minimum ventilation position.

3.7.7.2 Stepped-Speed Case

For the stepped-speed case, the RTU fan was modeled with four fan speed steps: off, minimum flow rate, reduced flow rate, and full flow rate. Minimum flow rate was used when the zone was in deadband operation, meaning that its HVAC system was not heating, cooling, or economizing. The minimum flow rates for both prototype models are detailed in Section 3.7.6. During Stage 1 cooling, the fan operated at a reduced flow rate equal to 80% of the design fan flow rate. This equated to 1.0 cfm/ft^2 (4.9 L/s·m^2) for the models with refrigeration and 0.8 cfm/ft^2 (4.1 L/s·m^2) for those without refrigeration. Full flow rate was used when the RTU was in heating, economizing, or Stage 2 cooling mode. The control logic narrative for the stepped-speed fan motor case follows.

1. Deadband
   a. During occupied hours, when the system was not in heating, economizing, or cooling mode, deadband mode was active.
      i. The fan flow rate was set to minimum. If this flow rate proved insufficient to maintain a supply air temperature (SAT) of at least 50°F, the fan flow rate was increased to 80% of the design fan flow rate. If this flow rate was still insufficient, the fan flow rate was increased to 100% of the design fan flow rate.
      ii. The OA dampers were modulated as appropriate to supply minimum ventilation.
      iii. The heating and cooling coils were turned off.
   b. During unoccupied hours, when the system was not in heating, economizing, or cooling mode, the RTU fan was turned off.
2. Heating
   a. The heating coil was engaged when the zone temperature was below the heating set point [$T_{zone} < T_{set,heat}$].
      i. The RTU fan flow rate was set equal to the design fan flow rate.
      ii. During occupied hours, the OA dampers were set to the minimum ventilation position.
      iii. During unoccupied hours, the OA dampers were closed.
      iv. The RTU continued to heat the zone until the zone temperature was at least 1°F above the heating set point [$T_{zone} \geq (T_{set,heat} + 1°F)$].
3. Economizing
   a. If (1) the OA temperature was less than 65°F and greater than 50°F (50°F < T_{OA} < 65°F), and (2) the zone temperature was above the cooling set point by no more than 1°F [$T_{set,cool} < T_{zone} \leq (T_{set,cool} + 1°F)$], economizing was engaged.
      i. The RTU fan flow rate was set equal to the design fan flow rate.
      ii. The OA dampers were set to 100% open.
      iii. The cooling coils remained off.

iv. The RTU operated in economizer mode until either:
    1. The OA temperature was less than or equal to 50°F ($T_{OA} \leq 50°F$), at which point the RTU entered Stage 1 cooling mode.
    2. The OA temperature was greater than or equal to 65°F ($T_{OA} \geq 65°F$), at which point the RTU entered Stage 1 cooling mode.
    3. The zone temperature was at least 1°F below the cooling set point [$T_{zone} \leq (T_{set,cool} - 1°F)$], at which point the RTU entered deadband mode.
    4. The zone temperature was greater than 1°F above the cooling set point [$T_{zone} > (T_{set,cool} + 1°F)$], at which point the RTU entered Stage 2 cooling mode.

4. Cooling
   a. Stage 1 cooling was engaged when the zone temperature rose above the cooling set point by no more than 1°F [$T_{set,cool} < T_{zone} \leq (T_{set,cool} + 1°F)$] and the OA temperature was not between 50°F and 65°F (economizing condition).
      i. The RTU fan flow rate was set equal to 80% of the design fan flow rate.
      ii. The first compressor was engaged.
      iii. The RTU remained in Stage 1 cooling until either:
         1. The zone temperature was at least 1°F below the cooling set point [$T_{zone} \leq (T_{set,cool} - 1°F)$], at which point the RTU entered deadband mode.
         2. The zone temperature was greater than 1°F above the cooling set point [$T_{zone} > (T_{set,cool} + 1°F)$], at which point the RTU entered Stage 2 cooling mode.
      iv. During unoccupied hours, the OA dampers were closed.
   b. Stage 2 cooling was engaged when the zone temperature was greater than 1°F above the cooling set point [$T_{zone} > (T_{set,cool} + 1°F)$].
      i. The RTU fan flow rate was set equal to the design fan flow rate.
      ii. Both compressors were engaged.
      iii. The RTU remained in Stage 2 cooling until the zone temperature was at least 1°F below the cooling set point [$T_{zone} \leq (T_{set,cool} - 1°F)$], at which point the RTU entered deadband mode.
      iv. During unoccupied hours, the OA dampers were closed, unless the OA temperature was less than 65°F and greater than 50°F ($50°F < T_{OA} < 65°F$), in which case the OA dampers were set to the minimum ventilation position.
   c. During occupied hours, the OA dampers were modulated to supply minimum ventilation.

3.7.7.3 Variable-Speed Case

For the variable-speed case, the fan flow rate was continuously varied, as opposed to being operated in discrete steps. Minimum flow rate was used when the zone was in deadband operation, meaning that its HVAC system was not heating, cooling, or economizing. The minimum flow rates for both model types are detailed in Section 3.7.6. During Stage 1 cooling, the fan flow rate was varied in order to hold the SAT at 55°F. Full flow rate was used when the RTU was in heating, economizing, or Stage 2 cooling mode. The control logic narrative for the variable-speed case follows.

1. Deadband
    a. During occupied hours, when the system was not in heating, economizing, or cooling mode, deadband mode was active.
        i. The fan flow rate was set to minimum. If this flow rate proved insufficient to maintain a SAT of at least 50°F, the RTU fan flow rate was varied between minimum and maximum flow to raise the SAT to 50°F.
        ii. The OA dampers were modulated as appropriate to supply minimum ventilation.
        iii. The heating and cooling coils were turned off.
    b. During unoccupied hours, when the system was not in heating, economizing, or cooling mode, the RTU fan was turned off.
2. Heating
    a. The heating coil was engaged when the zone temperature was below the heating set point [$T_{zone} < T_{set,heat}$].
        i. The RTU fan flow rate was set equal to the design fan flow rate.
        ii. During occupied hours, the OA dampers were set to the minimum ventilation position.
        iii. During unoccupied hours, the OA dampers were closed.
        iv. The RTU continued to heat the zone until the zone temperature was at least 1°F above the heating set point [$T_{zone} \geq (T_{set,heat} + 1°F)$].
3. Economizing
    a. If (1) the OA temperature was less than 65°F and greater than 50°F ($50°F < T_{OA} < 65°F$), and (2) the zone temperature was above the cooling set point by no more than 1°F [$T_{set,cool} < T_{zone} \leq (T_{set,cool} + 1°F)$], economizing was engaged.
        i. The RTU fan flow rate was set equal to the design fan flow rate.
        ii. The OA dampers were set to 100% open.
        iii. The cooling coils remained off.
        iv. The RTU operated in economizer mode until either:
            1. The OA temperature was less than or equal to 50°F ($T_{OA} \leq 50°F$), at which point the RTU entered Stage 1 cooling mode.
            2. The OA temperature was greater than or equal to 65°F ($T_{OA} \geq 65°F$), at which point the RTU entered Stage 1 cooling mode.
            3. The zone temperature was at least 1°F below the cooling set point [$T_{zone} \leq (T_{set,cool} - 1°F)$], at which point the RTU entered deadband mode.
            4. The zone temperature was greater than 1°F above the cooling set point [$T_{zone} > (T_{set,cool} + 1°F)$], at which point the RTU entered Stage 2 cooling mode.
4. Cooling
    a. Stage 1 cooling was engaged when the zone temperature rose above the cooling set point by no more than 1°F [$T_{set,cool} < T_{zone} \leq (T_{set,cool} + 1°F)$] and the OA temperature was not between 50°F and 65°F (economizing condition).
        i. The RTU fan flow rate was modulated between minimum and maximum flow, such that the SAT was equal to 55°F.
        ii. The first compressor was engaged.
        iii. The RTU remained in Stage 1 cooling until either:

1. The zone temperature was at least 1°F below the cooling set point [$T_{zone} \leq (T_{set,cool} - 1°F)$], at which point the RTU entered deadband mode.
2. The zone temperature was greater than 1°F above the cooling set point [$T_{zone} > (T_{set,cool} + 1°F)$], at which point the RTU entered Stage 2 cooling mode.

iv. During unoccupied hours, the OA dampers were closed.

b. Stage 2 cooling was engaged when the zone temperature was greater than 1°F above the cooling set point [$T_{zone} > (T_{set,cool} + 1°F)$].

i. The RTU fan flow rate was set equal to the design fan flow rate.
ii. Both compressors were engaged.
iii. The RTU remained in Stage 2 cooling until the zone temperature was at least 1°F below the cooling set point [$T_{zone} \leq (T_{set,cool} - 1°F)$], at which point the RTU entered deadband mode.
iv. During unoccupied hours, the OA dampers were closed, unless the OA temperature was less than 65°F and greater than 50°F ($50°F < T_{OA} < 65°F$), in which case the OA dampers were set to the minimum ventilation position.

c. During occupied hours, the OA dampers were modulated to supply minimum ventilation.

4.0 Results

This section describes the simulation results[3], including fan energy use, whole-building energy use, electricity consumption, natural gas consumption, utility cost savings, and observed relative humidity (RH) values.

4.1 Fan Energy Savings

Table 4–1 and Table 4–2 detail the predicted fan energy savings by climate zone, using total fan energy use intensity (EUI) as the basis for the savings calculations. As expected, the stepped-speed cases show significant energy savings over the constant-speed cases; the variable-speed cases show even greater savings in all climates.

Baseline fan energy use is constant across climate zones, because the supply fans were hard-sized using a climate-independent methodology to represent typical RTU fan over sizing (see Section 3.7.4).

Table 4–1 Normalized Annual Fan Energy Consumption for Models With Refrigeration

Climate Zone	Location	Constant-Speed Fan	Stepped-Speed Fan		Variable-Speed Fan	
		Fan EUI ($kBtu/ft^2$)	Fan EUI ($kBtu/ft^2$)	Savings (%)	Fan EUI ($kBtu/ft^2$)	Savings (%)
1A	Miami, FL	9.0	6.4	28.9	5.0	44.3
2A	Houston, TX	9.0	5.6	37.3	4.7	47.7
2B	Phoenix, AZ	9.0	5.6	37.5	4.7	48.0
3A	Atlanta, GA	9.0	5.0	44.4	4.2	53.1
3B	Los Angeles, CA	9.0	4.6	49.0	4.1	54.6
3B	Las Vegas, NV	9.0	5.3	41.4	4.5	50.4
3C	San Francisco, CA	9.0	4.0	55.1	3.9	56.2
4A	Baltimore, MD	9.0	4.9	45.7	4.3	51.9
4B	Albuquerque, NM	9.0	4.8	46.6	4.2	53.7
4C	Seattle, WA	9.0	4.2	52.9	4.1	54.5
5A	Chicago, IL	9.0	4.8	46.6	4.3	51.9
5B	Boulder, CO	9.0	4.6	48.3	4.2	52.7
6A	Minneapolis, MN	9.0	4.9	44.9	4.5	49.9
6B	Helena, MT	9.0	4.6	48.4	4.4	51.4
7	Duluth, MN	9.0	4.8	46.7	4.6	48.9
8	Fairbanks, AK	9.0	5.3	41.6	5.1	43.8

[3] Note that additional differences between the models with and without refrigeration, such as zoning layout and the fan flow rate sizing methodology, affect the energy consumption and savings values. Readers should exercise caution when comparing results between model types.

Table 4–2 Normalized Annual Fan Energy Consumption for Models Without Refrigeration

Climate Zone	Location	Constant-Speed Fan	Stepped-Speed Fan		Variable-Speed Fan	
		Fan EUI ($kBtu/ft^2$)	Fan EUI ($kBtu/ft^2$)	Savings (%)	Fan EUI ($kBtu/ft^2$)	Savings (%)
1A	Miami, FL	7.5	5.2	31.0	3.6	52.2
2A	Houston, TX	7.5	4.1	44.6	3.0	59.7
2B	Phoenix, AZ	7.5	4.1	45.6	3.1	58.8
3A	Atlanta, GA	7.5	3.4	54.1	2.5	66.7
3B	Los Angeles, CA	7.5	2.7	63.2	2.3	69.4
3B	Las Vegas, NV	7.5	3.6	51.5	2.8	63.1
3C	San Francisco, CA	7.5	1.9	74.3	1.9	75.0
4A	Baltimore, MD	7.5	3.4	54.6	2.5	65.9
4B	Albuquerque, NM	7.5	3.3	56.3	2.7	64.1
4C	Seattle, WA	7.5	2.3	69.7	2.1	71.7
5A	Chicago, IL	7.5	3.5	53.0	2.6	65.0
5B	Boulder, CO	7.5	3.2	57.6	2.6	64.7
6A	Minneapolis, MN	7.5	3.8	49.3	3.0	60.4
6B	Helena, MT	7.5	3.2	56.7	2.7	63.7
7	Duluth, MN	7.5	3.7	51.2	3.0	59.9
8	Fairbanks, AK	7.5	4.2	44.5	3.7	50.7

4.2 Whole-Building Energy Savings

Figure 4–1 through Figure 4–4 provide whole-building energy consumption comparisons for the stepped- and variable-speed cases, using total building EUI as the basis for the savings calculations. The figures also provide a breakdown of EUI by end use. Appendix B.1 includes tables detailing these data. Annual whole-building energy savings ranged from 0.7%–8.4% and were seen in all climate zones when switching to stepped- or variable-speed fan motor functionality.

Stepped- or variable-speed fan motor functionality decreased fan energy use in all climates, but also increased heating energy use. This outcome is attributed to the need to make up for beneficial fan heat that is no longer imparted to the supply airstream during the heating season. For the stepped- and variable-speed cases, the fans are operated at full flow rate during heating mode, so this heat is lost primarily when the systems are in deadband operation and the fans are operating at their minimum flow rate.

Cooling energy does not uniformly increase or decrease as a result of switching to stepped- or variable-speed fan motor functionality; however, there is a strong correlation between an increase in cooling energy and a decrease in average space RH (see Section 4.6). This is to be expected. As the cooling coil face velocity is decreased because of a reduction in flow rate during Stage 1 cooling for the stepped- and variable-speed cases, the cooling coils have more time to remove moisture from the airstream. This improvement in moisture removal effectiveness means more latent heat is removed from the airstream and subsequently more cooling energy is required to meet the zone temperature set point. For all locations, however, any increase in heating and cooling energy is always more than offset by the reduction in fan energy.

As shown in Table 4–3, whole-building energy savings vary from climate zone to climate zone when switching to stepped- or variable-speed fan motor functionality. Table 4–3 also shows that, when comparing the stepped- and the variable-speed cases, whole-building energy savings do not vary in a similar manner. For example, the whole-building energy savings observed in Seattle, WA, for the stepped- and variable speed models without refrigeration are almost identical, whereas there is a very large energy savings difference between these two models in Phoenix, AZ.

These trends are attributable to two factors: (1) climatic differences and (2) differences in the fan control logic implemented in each case. For example, in Seattle, WA, both models would be expected to operate in deadband or economizing mode the majority of the year. In deadband mode, the RTU supply fans operate at their minimum flow rates in both the stepped- and variable-speed cases. In economizing mode, the fans in both cases operate at their design flow rates. Because the fans in both models would be expected to operate in an identical manner for the majority of the year, the magnitude of the whole-building energy savings would be expected to be similar.

In contrast, for Phoenix, AZ, the models would likely spend much of their time in Stage 1 cooling. Because the fan flow rates are reduced in both the stepped- and variable-speed cases during Stage 1 cooling, energy savings would be expected in both cases. However, because the variable-speed control logic allows a greater possible reduction in fan flow rate, the whole-building energy savings for the variable-speed model would be expected to be greater than the savings for the stepped-speed model.

Examination of climate and associated impacts on RTU fan operation, as dictated by the case-specific control logic, can be used to explain the energy savings differences observed between the stepped- and variable-speed cases for the remaining climate zones.

Table 4–3 Annual Whole-Building Energy Savings for All Models (%)

Climate Zone	Location	Models With Refrigeration		Models Without Refrigeration	
		Stepped-Speed	Variable-Speed	Stepped-Speed	Variable-Speed
1A	Miami, FL	2.5	3.5	2.3	2.3
2A	Houston, TX	3.2	4.0	3.6	3.6
2B	Phoenix, AZ	3.5	4.7	4.1	5.2
3A	Atlanta, GA	3.2	3.9	3.7	3.7
3B	Los Angeles, CA	6.1	6.5	8.1	8.3
3B	Las Vegas, NV	3.4	4.6	4.1	5.5
3C	San Francisco, CA	5.8	5.9	8.3	8.4
4A	Baltimore, MD	2.2	2.6	2.5	2.4
4B	Albuquerque, NM	3.1	3.8	3.6	4.1
4C	Seattle, WA	3.1	3.2	4.0	4.0
5A	Chicago, IL	1.8	2.1	2.1	2.0
5B	Boulder, CO	2.6	3.1	3.2	3.5
6A	Minneapolis, MN	1.5	1.8	1.7	1.7
6B	Helena, MT	1.8	2.1	2.3	2.4
7	Duluth, MN	1.1	1.2	1.4	1.3
8	Fairbanks, AK	0.7	0.7	0.9	0.9

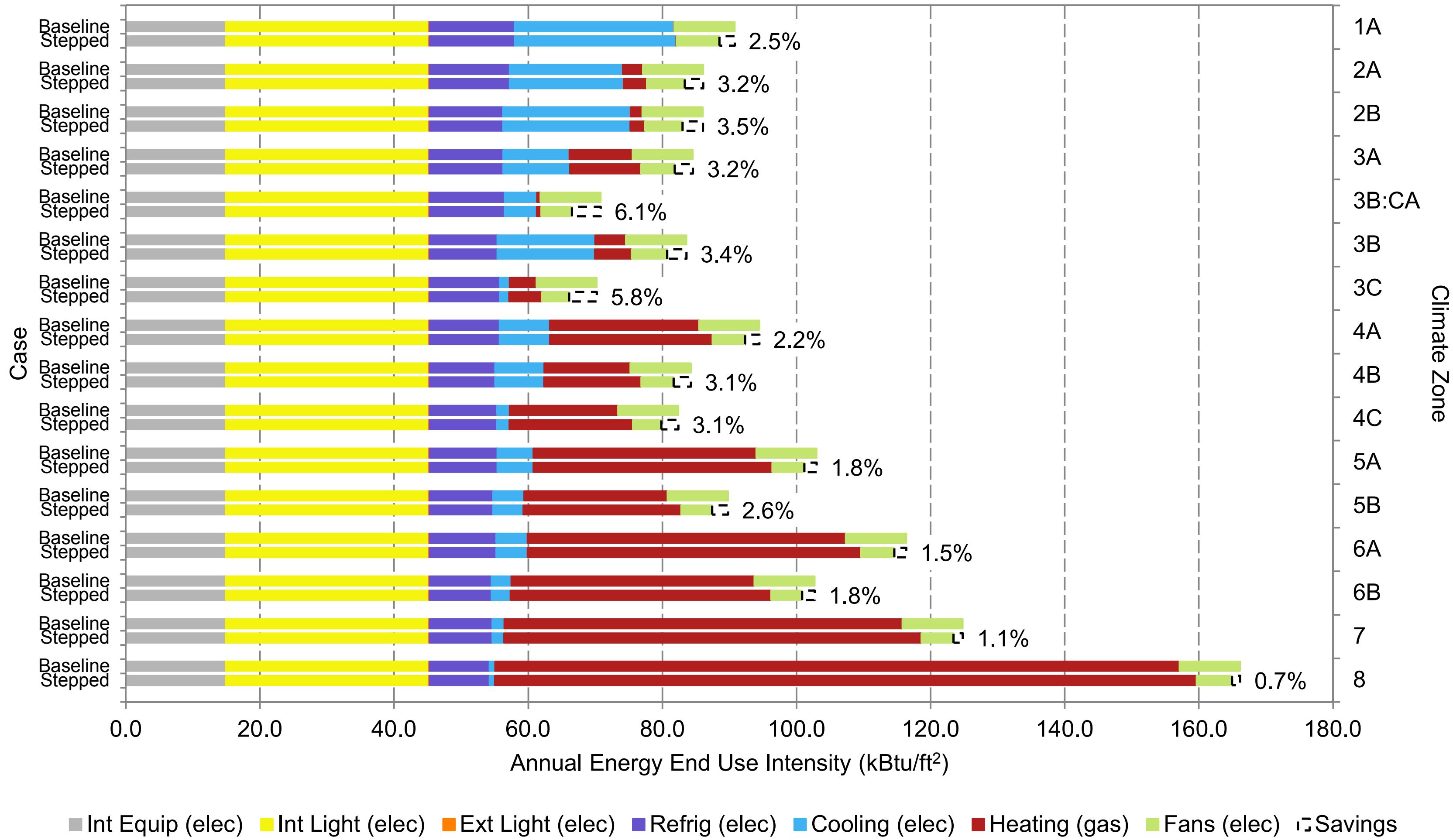

Figure 4–1 Models with refrigeration: annual whole-building EUI by end use for stepped-speed cases
(Credit: Lesley Herrmann/NREL)

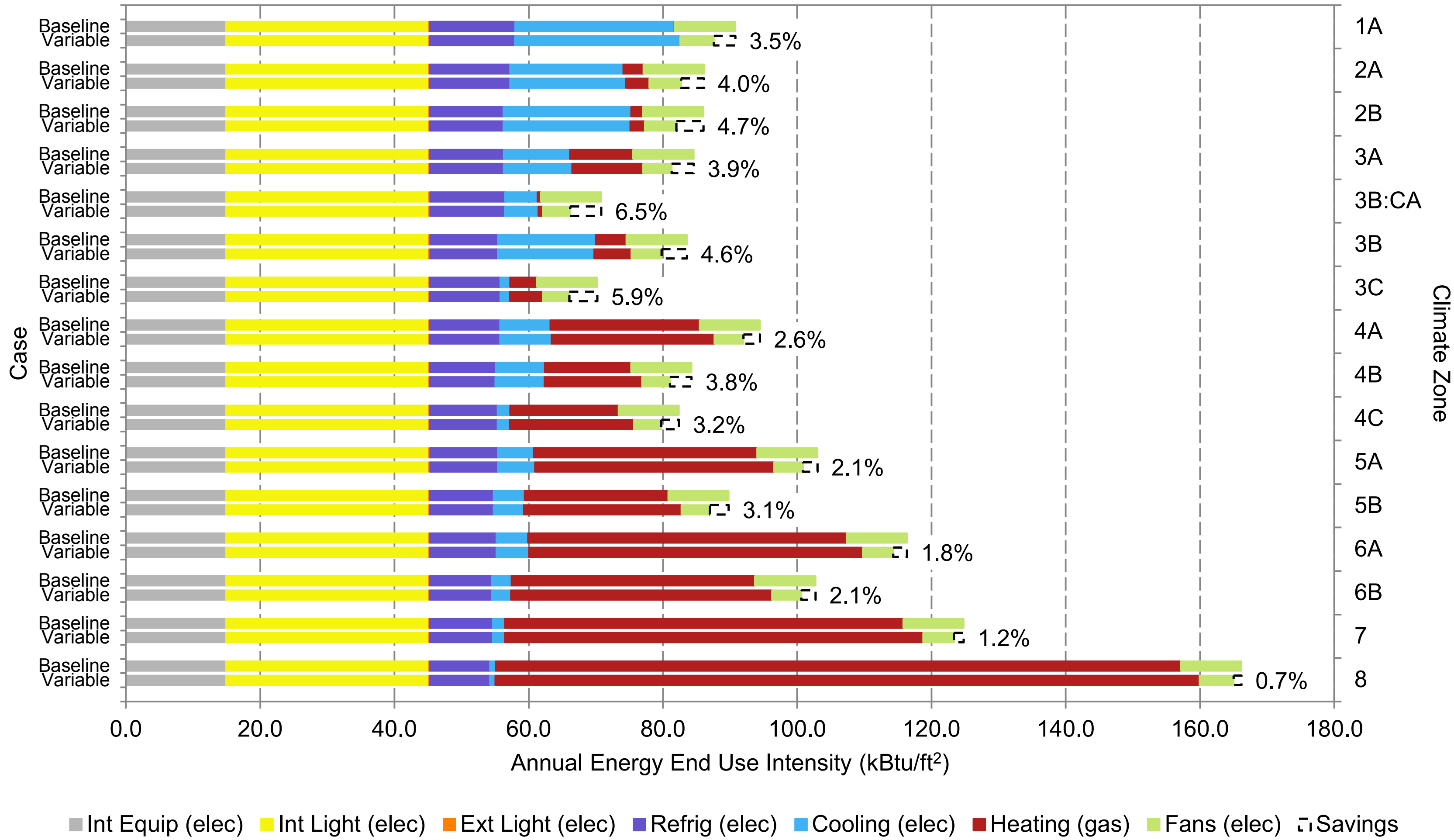

Figure 4–2 Models with refrigeration: annual whole-building EUI by end use for variable-speed cases
(Credit: Lesley Herrmann/NREL)

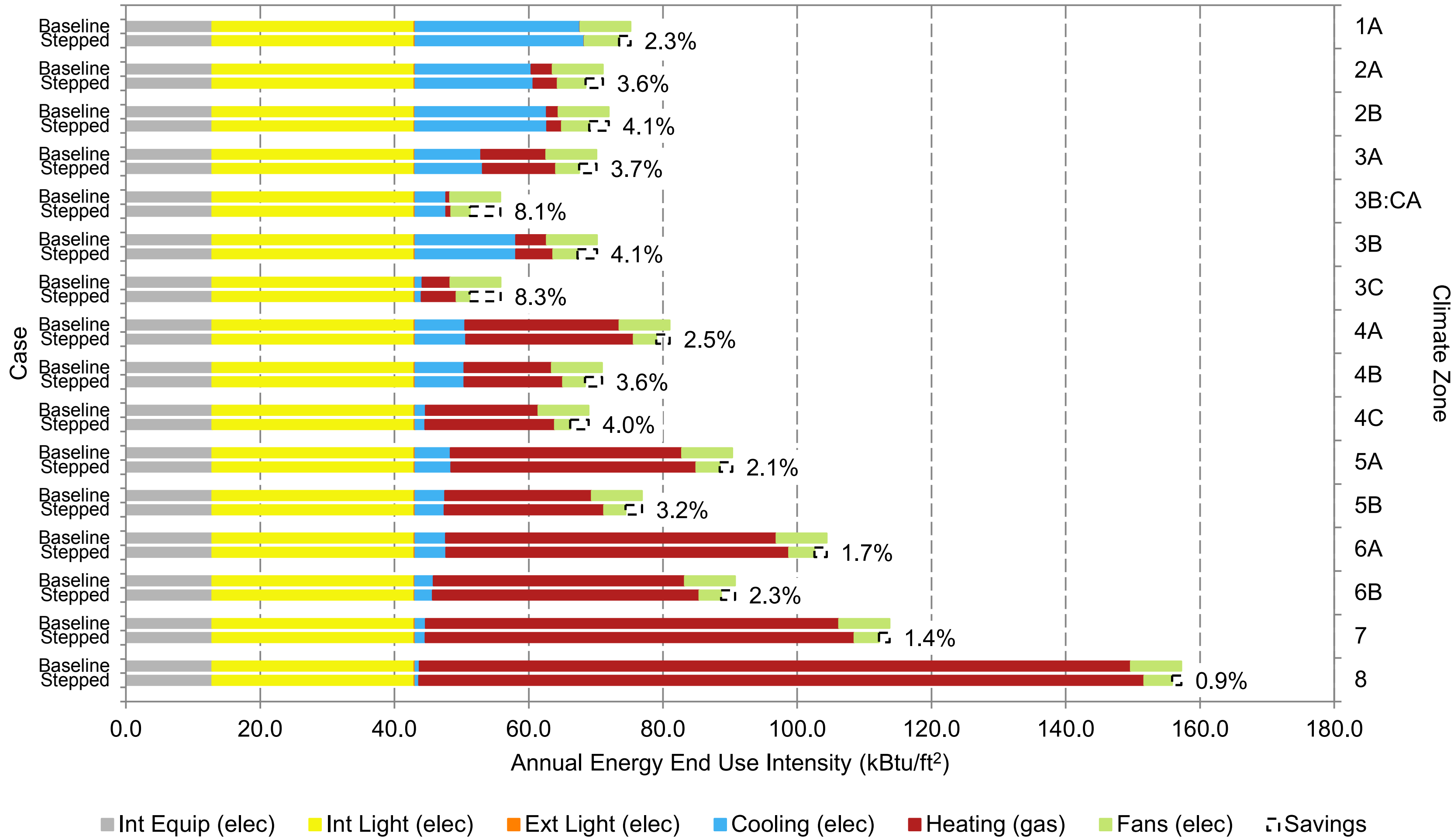

Figure 4–3 Models without refrigeration: annual whole-building EUI by end use for stepped-speed cases
(Credit: Lesley Herrmann/NREL)

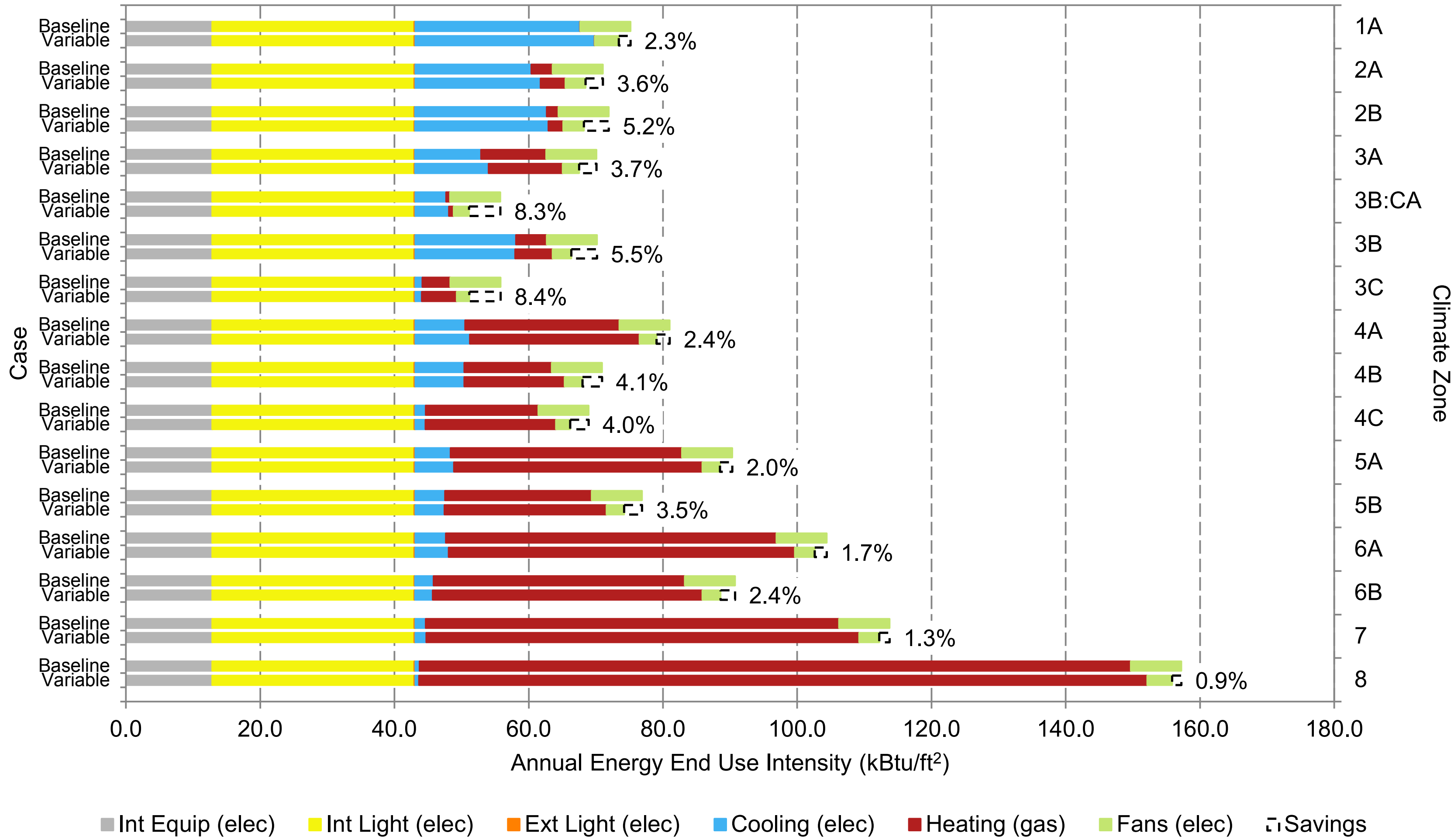

Figure 4–4 Models without refrigeration: annual whole-building EUI by end use for variable-speed cases
(Credit: Lesley Herrmann/NREL)

4.3 Electricity Consumption Savings

Table 4–4 and Table 4–5 detail annual whole-building electricity consumption, normalized by floor area, for all models. The stepped- and variable-speed cases show less electricity consumption than do the constant-speed fan motor cases; the variable-speed cases outperform the stepped-speed cases in every location.

The electricity savings presented in Table 4–4 and Table 4–5 appear to be small when normalized by area, but whole-building electricity savings may be substantial given the size of commercial buildings. Annual electricity savings can be estimated for a specific building by multiplying the normalized electricity savings by the building floor area. Annual electricity savings of up to 200,000 kWh are predicted for the models used in this study.

Sector-wide savings potential can be approximated by averaging the savings estimates from the 16 locations. For stepped-speed retrofits, the average savings was 1.17 kWh/ft^2; for variable-speed retrofits, it was 1.23 kWh/ft^2. If these savings were realized across 10% of the 2.7 billion ft^2 of retail space served by packaged RTUs (EIA 2008), retailers would save 316–332 GWh/yr (1.08–1.13 × 10^{12} Btu/yr).

Table 4–4 Normalized Annual Whole-Building Electricity Consumption for Models With Refrigeration

Climate Zone	Location	Constant-Speed Fan	Stepped-Speed Fan		Variable-Speed Fan	
		Use (kWh/ft^2)	Use (kWh/ft^2)	Savings (kWh/ft^2)	Use (kWh/ft^2)	Savings (kWh/ft^2)
1A	Miami, FL	26.51	25.85	0.66	25.57	0.94
2A	Houston, TX	24.28	23.33	0.95	23.15	1.13
2B	Phoenix, AZ	24.62	23.62	1.00	23.32	1.30
3A	Atlanta, GA	21.96	20.82	1.14	20.66	1.30
3B	Los Angeles, CA	20.55	19.24	1.31	19.15	1.40
3B	Las Vegas, NV	23.08	21.97	1.11	21.69	1.39
3C	San Francisco, CA	19.35	17.87	1.47	17.85	1.50
4A	Baltimore, MD	21.10	19.90	1.19	19.79	1.31
4B	Albuquerque, NM	20.86	19.62	1.24	19.43	1.43
4C	Seattle, WA	19.35	17.94	1.41	17.91	1.44
5A	Chicago, IL	20.38	19.16	1.22	19.06	1.32
5B	Boulder, CO	19.98	18.67	1.30	18.55	1.42
6A	Minneapolis, MN	20.13	18.95	1.18	18.85	1.28
6B	Helena, MT	19.41	18.11	1.30	18.03	1.39
7	Duluth, MN	19.12	17.88	1.24	17.84	1.29
8	Fairbanks, AK	18.73	17.61	1.12	17.55	1.18

Table 4–5 Normalized Annual Whole-Building Electricity Consumption for Models Without Refrigeration

Climate Zone	Location	Constant-Speed Fan	Stepped-Speed Fan		Variable-Speed Fan	
		Use (kWh/ft^2)	Use (kWh/ft^2)	Savings (kWh/ft^2)	Use (kWh/ft^2)	Savings (kWh/ft^2)
1A	Miami, FL	21.95	21.45	0.50	21.44	0.51
2A	Houston, TX	19.83	18.94	0.89	18.93	0.91
2B	Phoenix, AZ	20.49	19.51	0.98	19.28	1.22
3A	Atlanta, GA	17.65	16.53	1.11	16.51	1.13
3B	Los Angeles, CA	16.13	14.74	1.38	14.72	1.41
3B	Las Vegas, NV	19.16	18.03	1.13	17.75	1.41
3C	San Francisco, CA	15.09	13.43	1.66	13.42	1.67
4A	Baltimore, MD	16.95	15.79	1.16	15.71	1.24
4B	Albuquerque, NM	16.91	15.68	1.23	15.51	1.40
4C	Seattle, WA	15.23	13.68	1.55	13.66	1.57
5A	Chicago, IL	16.31	15.18	1.14	15.04	1.27
5B	Boulder, CO	16.08	14.79	1.29	14.64	1.44
6A	Minneapolis, MN	16.11	15.05	1.06	14.91	1.20
6B	Helena, MT	15.57	14.30	1.27	14.15	1.43
7	Duluth, MN	15.23	14.10	1.13	13.95	1.29
8	Fairbanks, AK	14.98	13.97	1.00	13.84	1.14

4.4 Natural Gas Consumption Savings

Table 4–6 and Table 4–7 detail whole-building natural gas consumption, normalized by floor area, for all models. Negative values indicate an increase in natural gas use. Because of the decrease in fan energy in the stepped- and variable-speed cases, the heating energy requirement increased, leading to an increase in natural gas use in all models.

Annual natural gas savings can be estimated for a specific building by multiplying the normalized natural gas savings contained in Table 4–6 and Table 4–7 by the building floor area. We performed this calculation and predict that the worst-case increase in natural gas use would be 3,900 therm/yr for the models used in this study.

On average, annual natural gas use increased by 0.0149 $therm/ft^2$ for the stepped-speed cases and by 0.0162 $therm/ft^2$ for the variable-speed cases. If these savings were realized across 10% of the 2.7 billion ft^2 of retail space served by packaged RTUs (EIA 2008), retailers would increase natural gas use by 4.0–4.4 million therm/yr (4.02–4.34 × 10^{11} Btu/yr).

Table 4–6 Normalized Annual Whole-Building Natural Gas Consumption* for Models With Refrigeration

Climate Zone	Location	Constant-Speed Fan	Stepped-Speed Fan		Variable-Speed Fan	
		Use (therm/ft^2)	Use (therm/ft^2)	Savings (therm/ft^2)	Use (therm/ft^2)	Savings (therm/ft^2)
1A	Miami, FL	0.000	0.000	0.000	0.000	0.000
2A	Houston, TX	0.030	0.034	–0.005	0.034	–0.005
2B	Phoenix, AZ	0.018	0.022	–0.004	0.022	–0.004
3A	Atlanta, GA	0.094	0.105	–0.012	0.105	–0.012
3B	Los Angeles, CA	0.005	0.006	–0.002	0.006	–0.002
3B	Las Vegas, NV	0.046	0.055	–0.009	0.055	–0.009
3C	San Francisco, CA	0.040	0.049	–0.009	0.049	–0.010
4A	Baltimore, MD	0.222	0.241	–0.020	0.242	–0.020
4B	Albuquerque, NM	0.128	0.145	–0.017	0.145	–0.017
4C	Seattle, WA	0.161	0.184	–0.023	0.184	–0.023
5A	Chicago, IL	0.331	0.355	–0.023	0.355	–0.024
5B	Boulder, CO	0.213	0.234	–0.021	0.234	–0.021
6A	Minneapolis, MN	0.473	0.496	–0.023	0.496	–0.023
6B	Helena, MT	0.361	0.387	–0.026	0.388	–0.026
7	Duluth, MN	0.591	0.620	–0.029	0.621	–0.029
8	Fairbanks, AK	1.017	1.043	–0.026	1.045	–0.028

*Negative values indicate that natural gas use increased.

Table 4–7 Normalized Annual Whole-Building Natural Gas Consumption* for Models Without Refrigeration

Climate Zone	Location	Constant-Speed Fan	Stepped-Speed Fan		Variable-Speed Fan	
		Use (therm/ft^2)	Use (therm/ft^2)	Savings (therm/ft^2)	Use (therm/ft^2)	Savings (therm/ft^2)
1A	Miami, FL	0.000	0.000	0.000	0.000	0.000
2A	Houston, TX	0.031	0.036	-0.005	0.037	-0.005
2B	Phoenix, AZ	0.018	0.022	-0.005	0.022	-0.005
3A	Atlanta, GA	0.097	0.109	-0.012	0.110	-0.013
3B	Los Angeles, CA	0.006	0.007	-0.002	0.007	-0.002
3B	Las Vegas, NV	0.045	0.055	-0.010	0.055	-0.010
3C	San Francisco, CA	0.041	0.052	-0.011	0.052	-0.011
4A	Baltimore, MD	0.229	0.249	-0.020	0.252	-0.023
4B	Albuquerque, NM	0.130	0.147	-0.017	0.149	-0.019
4C	Seattle, WA	0.167	0.193	-0.025	0.193	-0.026
5A	Chicago, IL	0.344	0.364	-0.020	0.369	-0.025
5B	Boulder, CO	0.218	0.237	-0.019	0.240	-0.023
6A	Minneapolis, MN	0.491	0.509	-0.018	0.514	-0.023
6B	Helena, MT	0.373	0.396	-0.023	0.400	-0.027
7	Duluth, MN	0.614	0.637	-0.023	0.643	-0.029
8	Fairbanks, AK	1.056	1.077	-0.021	1.081	-0.025

*Negative values indicate that natural gas use increased.

4.5 Utility Cost Savings

The energy savings values presented here depend on a number of assumptions, including the cooling coil and fan flow rate sizing methodology, sequence of operations, space set points, and hours of operation. In order to provide broadly applicable findings, values were used that reflect common practice.

Appendix C provides instructions about how to use the results of this study to estimate high-level utility cost savings. A retailer can use such results to justify site-specific assessments of retrofit feasibility and determine if a compelling business case can be made.

Assuming typical utility rates of \$0.10/kWh and \$1.00/therm, if average savings were realized across 10% of the 2.7 billion ft^2 of retail space served by packaged RTUs (EIA 2008), retailers would save \$27.6 million annually with stepped-speed retrofits, or \$28.8 million with variable-speed retrofits.

4.6 Relative Humidity

Table 4–8 and Table 4–9 detail the 95th percentile RH values for all models. These values were chosen to exclude extreme outliers, but still provide an understanding of space maximum RH. RH values are reported for the "grocery" zone for the models with refrigeration and for the "sales 1" zone for the models without refrigeration, as these represented the critical zones for each building type.

Active zone RH control was outside the scope of this study, and the lack of such control explains the high RH values observed in most models, particularly those seen in the humid "A" climate subzones. The models captured the effect on space RH caused by moisture condensing on the cooling coil surface as the coil met the zone sensible load. The models did not, however, account for the re-entrainment of moisture into the airstream, which would have occurred in practice when the cooling coil was shut off and condensation on the cooling coil evaporated back into the airstream. Thus, the negative impact on space RH caused by frequent cooling coil cycling (and conversely, the RH benefit of switching to stepped- or variable-speed fan motor functionality) was not fully captured. Therefore, RH improvements observed in most climate zones would likely be even greater in practice.

The reported RH values for the stepped- and variable-speed cases increase minimally in climate zones 3C and 8. These increases are negligible and are due to very slight differences in the zone dry-bulb temperatures between cases.

Table 4–8 Models With Refrigeration: 95th Percentile RH Values

Climate Zone	Location	Constant-Speed Fan RH (%)	Stepped-Speed Fan RH (%)	Variable-Speed Fan RH (%)
1A	Miami, FL	72.5	71.9	70.2
2A	Houston, TX	71.8	71.2	69.9
2B	Phoenix, AZ	56.2	55.5	53.8
3A	Atlanta, GA	66.7	66.2	64.5
3B	Los Angeles, CA	59.2	58.9	58.0
3B	Las Vegas, NV	42.2	42.0	42.0
3C	San Francisco, CA	49.4	49.6	49.5
4A	Baltimore, MD	67.8	67.3	65.6
4B	Albuquerque, NM	48.3	47.9	46.7
4C	Seattle, WA	50.8	50.9	50.3
5A	Chicago, IL	63.1	62.7	61.1
5B	Boulder, CO	46.8	46.5	45.8
6A	Minneapolis, MN	63.0	62.7	61.0
6B	Helena, MT	40.1	40.1	40.1
7	Duluth, MN	56.9	56.5	55.8
8	Fairbanks, AK	45.5	45.8	45.8

Table 4–9 Models Without Refrigeration: 95th Percentile RH Values

Climate Zone	Location	Constant-Speed Fan RH (%)	Stepped-Speed Fan RH (%)	Variable-Speed Fan RH (%)
1A	Miami, FL	73.2	72.4	69.5
2A	Houston, TX	72.9	72.4	69.7
2B	Phoenix, AZ	56.3	55.4	53.1
3A	Atlanta, GA	69.2	68.0	63.7
3B	Los Angeles, CA	63.2	62.6	60.7
3B	Las Vegas, NV	45.8	45.7	45.3
3C	San Francisco, CA	54.2	54.5	54.2
4A	Baltimore, MD	70.1	69.2	65.5
4B	Albuquerque, NM	51.7	51.0	49.7
4C	Seattle, WA	55.6	55.5	54.8
5A	Chicago, IL	66.4	65.5	61.3
5B	Boulder, CO	49.8	49.3	48.7
6A	Minneapolis, MN	67.1	65.9	62.0
6B	Helena, MT	44.3	44.2	44.2
7	Duluth, MN	62.1	61.7	59.0
8	Fairbanks, AK	50.2	50.4	50.3

5.0 Conclusions

Our simulations showed annual whole-building and electric energy savings in all climate zones when constant-speed RTU fan motors were upgraded with stepped- or variable-speed functionality. Whole-building energy savings ranged from 0.7%–8.4%.

Normalized electricity savings ranged from 0.50–1.67 kWh/ft^2 (1.70–5.70 kBtu/ft^2). Average annual electricity savings across all locations were 1.17 kWh/ft^2 (3.98 kBtu/ft^2) for the stepped-speed cases and 1.23 kWh/ft^2 (4.40 kBtu/ft^2) for the variable-speed cases. The results suggest that for certain building types, annual whole-building electricity savings of 200,000 kWh may be realized.

Heating energy increased in all climates, because less fan energy was imparted to the supply airstream during the heating season. On average, this increase was 0.0149 therm/ft^2 for the stepped-speed cases and 0.0162 therm/ft^2 for the variable-speed cases. (This increase was always more than offset by the electricity use reduction.)

Based on the average energy savings, if variable-speed retrofits were implemented across 10% of the 2.7 billion ft^2 of retail space served by packaged RTUs (EIA 2008), retailers would save approximately 332 GWh/yr (1.13×10^{12} Btu/yr) of electricity and increase natural gas use by 4.4 million therm/yr (4.34×10^{11} Btu/yr). Assuming typical utility rates of $0.10/kWh and $1.00/therm, this would equate to annual utility cost savings of $28.8 million.

This measure requires that a relatively small number of RTU fan motors be upgraded per building. Therefore, this report makes a case that such retrofits should be investigated in all climate zones to address the oftentimes dispersed energy loads in commercial buildings.

6.0 Next Steps

The energy savings values presented here depend on a number of assumptions, including the cooling coil and fan flow rate sizing methodology, sequence of operations, space set points, and hours of operation. In order to provide broadly applicable findings, values were used that reflect common practice.

Appendix C provides instructions about how to use the results of this study to estimate high-level utility cost savings. A retailer can use such results to justify site-specific assessments of retrofit feasibility and determine if a compelling business case can be made.

Indirect benefits may be sufficient to motivate retailers to pursue constant-speed RTU fan motor retrofits, especially for those who own or operate buildings in humid regions. Because moisture problems are largely attributed to short-cycling of oversized HVAC equipment, future work could focus on enhancing the EnergyPlus whole-building energy models to predict the effects of stepped- and variable-speed fan motor retrofits on moisture re-entrainment. For a retail building with moisture control issues, such modeling could yield recommendations about whether fan motor retrofits would significantly improve moisture control. Simulation would enable this analysis to span a range of climates and building types.

Physical testing could also improve the assessment of associated benefits. Whereas simulation can examine retrofit impacts over a wide range of conditions, carefully designed experimental testing could uncover important real-world considerations. Testing results could also be used to validate and improve the numerical models used for this study. By retrofitting a constant-speed RTU with variable-speed functionality, stepped- and variable-speed performance could be determined empirically to complement or refine this analysis. Using such techniques to capture moisture control advantages, component-specific impacts, and other drivers may accelerate large-scale retailer investment in energy-saving RTU fan motor retrofits.

The development of an application guide to help retailers effectively execute stepped- and variable-speed RTU fan motor retrofits would also help to accelerate adoption of this technology. Ideally, such a guide would provide high-level implementation advice and be paired with manufacturer-created retrofit packages that would facilitate retrofit installation while maintaining the original equipment warranty. The development of such a solution would require involvement from RTU manufacturers, private sector retailers, and research and development organizations. While the level of collaboration required to successfully execute these tasks is significant, the potential benefits associated with this kind of a documentation/product solution pair warrant a closer examination of this path.

7.0 References

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Appendix A Schedules

A.1 Occupancy, Lighting, and Equipment Schedules

Occupancy, lighting, and equipment schedules are presented in Table A–1 and Figure A–1.

Table A–1 Occupancy, Lighting, and Equipment Schedules

Hour	Occupancy (Percent of Peak)	Lighting (Percent of Peak)	Plug Loads (Percent of Peak)
1	0	5	40
2	0	5	40
3	0	5	40
4	0	5	40
5	0	5	40
6	0	5	40
7	0	5	40
8	30	25	95
9	50	100	95
10	70	100	95
11	90	100	95
12	90	100	95
13	90	100	95
14	70	100	95
15	70	100	95
16	70	100	95
17	90	100	95
18	90	100	95
19	70	100	95
20	70	100	95
21	50	100	95
22	50	100	95
23	30	100	95
24	0	25	40

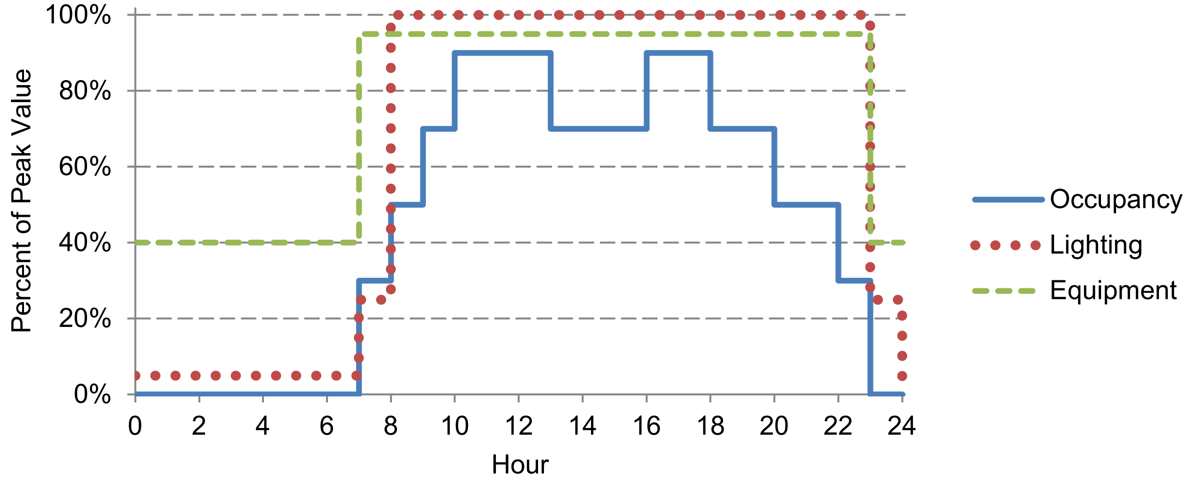

Figure A–1 Occupancy, lighting, and equipment schedules
(Credit: Rachel Romero/NREL)

A.2 Heating and Cooling Set Point Schedules

Figure A–2 and Table A–2 detail the heating and cooling set point values for each hour of the day. Set point temperatures during occupied hours were 68°F (heating) and 74°F (cooling). During unoccupied hours, the set point temperatures were 59°F (heating) and 81°F (cooling). These set point schedules were applied seven days per week, year-round.

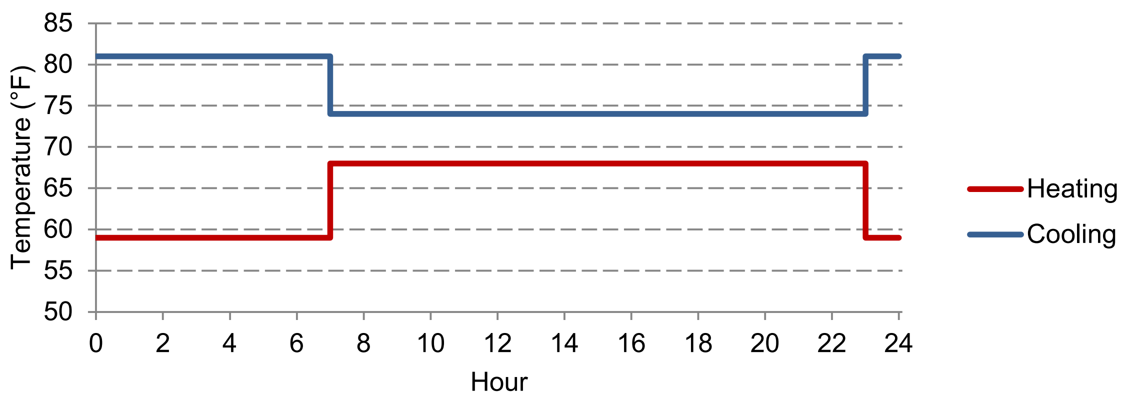

Figure A–2 Heating and cooling set points
(Credit: Rachel Romero/NREL)

Table A–2 Heating and Cooling Set Points

Hour	Heating (°F)	Heating (°C)	Cooling (°F)	Cooling (°C)
1	59.0	15.0	81.0	27.2
2	59.0	15.0	81.0	27.2
3	59.0	15.0	81.0	27.2
4	59.0	15.0	81.0	27.2
5	59.0	15.0	81.0	27.2
6	59.0	15.0	81.0	27.2
7	59.0	15.0	81.0	27.2
8	68.0	20.0	74.0	23.3
9	68.0	20.0	74.0	23.3
10	68.0	20.0	74.0	23.3
11	68.0	20.0	74.0	23.3
12	68.0	20.0	74.0	23.3
13	68.0	20.0	74.0	23.3
14	68.0	20.0	74.0	23.3
15	68.0	20.0	74.0	23.3
16	68.0	20.0	74.0	23.3
17	68.0	20.0	74.0	23.3
18	68.0	20.0	74.0	23.3
19	68.0	20.0	74.0	23.3
20	68.0	20.0	74.0	23.3
21	68.0	20.0	74.0	23.3
22	68.0	20.0	74.0	23.3
23	59.0	15.0	81.0	27.2
24	59.0	15.0	81.0	27.2

Appendix B Additional Results Tables

B.1 Detailed Whole-Building Results

Table B–1 Energy End Use Breakdown for Models With Refrigeration – IP Units

Climate Zone	Location	Model	Gas ($kBtu/ft^2 \cdot yr$) Heating	Electricity ($kBtu/ft^2 \cdot yr$) Cooling	Fans	Interior Lights	Exterior Lights	Equipment	Refrigeration	Total ($kBtu/ft^2 \cdot yr$)	Savings (%)
1A	Miami, FL	Constant	0.0	23.7	9.0	30.2	0.1	14.9	12.6	90.5	–
		Stepped	0.0	24.0	6.4	30.2	0.1	14.9	12.6	88.2	2.5
		Variable	0.0	24.5	5.0	30.2	0.1	14.9	12.6	87.3	3.5
2A	Houston, TX	Constant	3.0	16.8	9.0	30.2	0.1	14.9	11.9	85.8	–
		Stepped	3.4	16.9	5.6	30.2	0.1	14.9	11.9	83.1	3.2
		Variable	3.4	17.2	4.7	30.2	0.1	14.9	11.9	82.4	4.0
2B	Phoenix, AZ	Constant	1.8	18.9	9.0	30.2	0.1	14.9	10.9	85.8	–
		Stepped	2.2	18.9	5.6	30.2	0.1	14.9	10.9	82.7	3.5
		Variable	2.2	18.8	4.7	30.2	0.1	14.9	10.9	81.7	4.7
3A	Atlanta, GA	Constant	9.4	9.8	9.0	30.2	0.1	14.9	11.0	84.3	–
		Stepped	10.5	9.9	5.0	30.2	0.1	14.9	11.0	81.6	3.2
		Variable	10.5	10.2	4.2	30.2	0.1	14.9	10.9	81.0	3.9
3B	Los Angeles, CA	Constant	0.5	4.8	9.0	30.2	0.1	14.9	11.1	70.6	–
		Stepped	0.6	4.8	4.6	30.2	0.1	14.9	11.1	66.3	6.1
		Variable	0.6	5.0	4.1	30.2	0.1	14.9	11.1	66.0	6.5
3B	Las Vegas, NV	Constant	4.6	14.5	9.0	30.2	0.1	14.9	10.1	83.3	–
		Stepped	5.5	14.5	5.3	30.2	0.1	14.9	10.1	80.5	3.4
		Variable	5.5	14.3	4.5	30.2	0.1	14.9	10.1	79.5	4.6
3C	San Francisco, CA	Constant	4.0	1.5	9.0	30.2	0.1	14.9	10.4	70.0	–
		Stepped	4.9	1.4	4.0	30.2	0.1	14.9	10.4	65.9	5.8
		Variable	4.9	1.4	3.9	30.2	0.1	14.9	10.4	65.8	5.9
4A	Baltimore, MD	Constant	22.2	7.5	9.0	30.2	0.1	14.9	10.4	94.2	–
		Stepped	24.1	7.5	4.9	30.2	0.1	14.9	10.4	92.1	2.2
		Variable	24.2	7.7	4.3	30.2	0.1	14.9	10.4	91.7	2.6

Table B–1 Energy End Use Breakdown for Models With Refrigeration – IP Units (Continued)

Climate Zone	Location	Model	Gas ($kBtu/ft^2 \cdot yr$)	Electricity ($kBtu/ft^2 \cdot yr$)						Total ($kBtu/ft^2 \cdot yr$)	Savings (%)
			Heating	Cooling	Fans	Interior Lights	Exterior Lights	Equipment	Refrigeration		
4B	Albuquerque, NM	Constant	12.8	7.3	9.0	30.2	0.1	14.9	9.7	84.0	–
		Stepped	14.5	7.3	4.8	30.2	0.1	14.9	9.7	81.4	3.1
		Variable	14.5	7.3	4.2	30.2	0.1	14.9	9.7	80.8	3.8
4C	Seattle, WA	Constant	16.1	1.9	9.0	30.2	0.1	14.9	10.0	82.1	–
		Stepped	18.4	1.8	4.2	30.2	0.1	14.9	10.0	79.6	3.1
		Variable	18.4	1.9	4.1	30.2	0.1	14.9	10.0	79.5	3.2
5A	Chicago, IL	Constant	33.1	5.4	9.0	30.2	0.1	14.9	10.0	102.7	–
		Stepped	35.5	5.4	4.8	30.2	0.1	14.9	10.0	100.8	1.8
		Variable	35.5	5.5	4.3	30.2	0.1	14.9	10.0	100.5	2.1
5B	Boulder, CO	Constant	21.3	4.6	9.0	30.2	0.1	14.9	9.4	89.5	–
		Stepped	23.4	4.5	4.6	30.2	0.1	14.9	9.4	87.2	2.6
		Variable	23.4	4.5	4.2	30.2	0.1	14.9	9.4	86.7	3.1
6A	Minneapolis, MN	Constant	47.3	4.7	9.0	30.2	0.1	14.9	9.9	116.0	–
		Stepped	49.6	4.7	4.9	30.2	0.1	14.9	9.9	114.2	1.5
		Variable	49.6	4.8	4.5	30.2	0.1	14.9	9.9	113.9	1.8
6B	Helena, MT	Constant	36.1	2.9	9.0	30.2	0.1	14.9	9.2	102.4	–
		Stepped	38.7	2.8	4.6	30.2	0.1	14.9	9.2	100.5	1.8
		Variable	38.8	2.8	4.4	30.2	0.1	14.9	9.2	100.3	2.1
7A	Duluth, MN	Constant	59.1	1.8	9.0	30.2	0.1	14.9	9.3	124.4	–
		Stepped	62.0	1.7	4.8	30.2	0.1	14.9	9.3	123.0	1.1
		Variable	62.1	1.8	4.6	30.2	0.1	14.9	9.3	122.9	1.2
8A	Fairbanks, AK	Constant	101.7	0.8	9.0	30.2	0.1	14.9	8.9	165.6	–
		Stepped	104.3	0.8	5.3	30.2	0.1	14.9	8.9	164.4	0.7
		Variable	104.5	0.8	5.1	30.2	0.1	14.9	8.9	164.4	0.7

Table B–2 Energy End Use Breakdown for Models Without Refrigeration – IP Units

Climate Zone	Location	Model	Gas ($kBtu/ft^2·yr$)	Electricity ($kBtu/ft^2·yr$)					Total ($kBtu/ft^2·yr$)	Savings (%)
			Heating	Cooling	Fans	Interior Lights	Exterior Lights	Equipment		
1A	Miami, FL	Constant	0.0	24.5	7.5	30.0	0.2	12.8	74.9	–
		Stepped	0.0	25.1	5.2	30.0	0.2	12.8	73.2	2.3
		Variable	0.0	26.6	3.6	30.0	0.2	12.8	73.2	2.3
2A	Houston, TX	Constant	3.1	17.3	7.5	30.0	0.2	12.8	70.8	–
		Stepped	3.6	17.5	4.1	30.0	0.2	12.8	68.2	3.6
		Variable	3.7	18.6	3.0	30.0	0.2	12.8	68.3	3.6
2B	Phoenix, AZ	Constant	1.8	19.5	7.5	30.0	0.2	12.8	71.7	–
		Stepped	2.2	19.6	4.1	30.0	0.2	12.8	68.8	4.1
		Variable	2.2	19.8	3.1	30.0	0.2	12.8	68.0	5.2
3A	Atlanta, GA	Constant	9.7	9.8	7.5	30.0	0.2	12.8	69.9	–
		Stepped	10.9	10.0	3.4	30.0	0.2	12.8	67.3	3.7
		Variable	11.0	10.9	2.5	30.0	0.2	12.8	67.3	3.7
3B	Los Angeles, CA	Constant	0.6	4.6	7.5	30.0	0.2	12.8	55.6	–
		Stepped	0.7	4.6	2.7	30.0	0.2	12.8	51.1	8.1
		Variable	0.7	5.0	2.3	30.0	0.2	12.8	51.0	8.3
3B	Las Vegas, NV	Constant	4.5	14.9	7.5	30.0	0.2	12.8	69.9	–
		Stepped	5.5	14.9	3.6	30.0	0.2	12.8	67.1	4.1
		Variable	5.5	14.8	2.8	30.0	0.2	12.8	66.1	5.5
3C	San Francisco, CA	Constant	4.1	1.1	7.5	30.0	0.2	12.8	55.6	–
		Stepped	5.2	0.9	1.9	30.0	0.2	12.8	51.0	8.3
		Variable	5.2	1.0	1.9	30.0	0.2	12.8	51.0	8.4
4A	Baltimore, MD	Constant	22.9	7.4	7.5	30.0	0.2	12.8	80.7	–
		Stepped	24.9	7.5	3.4	30.0	0.2	12.8	78.8	2.5
		Variable	25.2	8.1	2.5	30.0	0.2	12.8	78.8	2.4

Table B–2 Energy End Use Breakdown for Models Without Refrigeration – IP Units (Continued)

Climate Zone	Location	Model	Gas ($kBtu/ft^2 \cdot yr$) Heating	Electricity ($kBtu/ft^2 \cdot yr$) Cooling	Fans	Interior Lights	Exterior Lights	Equipment	Total ($kBtu/ft^2 \cdot yr$)	Savings (%)
4B	Albuquerque, NM	Constant	13.0	7.3	7.5	30.0	0.2	12.8	70.7	–
		Stepped	14.7	7.3	3.3	30.0	0.2	12.8	68.1	3.6
		Variable	14.9	7.3	2.7	30.0	0.2	12.8	67.8	4.1
4C	Seattle, WA	Constant	16.7	1.5	7.5	30.0	0.2	12.8	68.7	–
		Stepped	19.3	1.5	2.3	30.0	0.2	12.8	66.0	4.0
		Variable	19.3	1.5	2.1	30.0	0.2	12.8	65.9	4.0
5A	Chicago, IL	Constant	34.4	5.2	7.5	30.0	0.2	12.8	90.0	–
		Stepped	36.4	5.3	3.5	30.0	0.2	12.8	88.2	2.1
		Variable	36.9	5.7	2.6	30.0	0.2	12.8	88.2	2.0
5B	Boulder, CO	Constant	21.8	4.4	7.5	30.0	0.2	12.8	76.6	–
		Stepped	23.7	4.3	3.2	30.0	0.2	12.8	74.2	3.2
		Variable	24.0	4.3	2.6	30.0	0.2	12.8	74.0	3.5
6A	Minneapolis, MN	Constant	49.1	4.5	7.5	30.0	0.2	12.8	104.0	–
		Stepped	50.9	4.6	3.8	30.0	0.2	12.8	102.2	1.7
		Variable	51.4	5.0	3.0	30.0	0.2	12.8	102.3	1.7
6B	Helena, MT	Constant	37.3	2.7	7.5	30.0	0.2	12.8	90.4	–
		Stepped	39.6	2.6	3.2	30.0	0.2	12.8	88.4	2.3
		Variable	40.0	2.6	2.7	30.0	0.2	12.8	88.3	2.4
7A	Duluth, MN	Constant	61.4	1.5	7.5	30.0	0.2	12.8	113.4	–
		Stepped	63.7	1.5	3.7	30.0	0.2	12.8	111.8	1.4
		Variable	64.3	1.6	3.0	30.0	0.2	12.8	111.8	1.3
8A	Fairbanks, AK	Constant	105.6	0.6	7.5	30.0	0.2	12.8	156.7	–
		Stepped	107.7	0.5	4.2	30.0	0.2	12.8	155.3	0.9
		Variable	108.1	0.6	3.7	30.0	0.2	12.8	155.3	0.9

Table B–3 Energy End Use Breakdown for Models With Refrigeration – SI Units

Climate Zone	Location	Model	Gas ($MJ/m^2 \cdot yr$) Heating	Electricity ($MJ/m^2 \cdot yr$) Cooling	Fans	Interior Lights	Exterior Lights	Equipment	Refrigeration	Total ($MJ/m^2 \cdot yr$)	Savings (%)
1A	Miami, FL	Constant	0.2	269.7	102.7	343.6	1.5	169.3	143.9	1,031.0	–
		Stepped	0.3	273.6	73.1	343.6	1.5	169.3	143.9	1,005.2	2.5
		Variable	0.3	279.0	57.2	343.6	1.5	169.3	143.6	994.5	3.5
2A	Houston, TX	Constant	33.9	191.6	102.5	343.6	1.5	169.3	135.5	977.9	–
		Stepped	39.1	193.0	64.3	343.6	1.5	169.3	135.5	946.3	3.2
		Variable	39.2	196.5	53.7	343.6	1.5	169.3	135.4	939.1	4.0
2B	Phoenix, AZ	Constant	20.0	215.8	102.5	343.6	1.5	169.3	124.3	977.1	–
		Stepped	24.5	215.4	64.1	343.6	1.5	169.3	124.3	942.8	3.5
		Variable	24.6	214.6	53.3	343.6	1.5	169.3	124.2	931.1	4.7
3A	Atlanta, GA	Constant	106.7	111.9	102.4	343.6	1.4	169.3	124.9	960.3	–
		Stepped	120.0	113.1	57.0	343.6	1.4	169.3	124.9	929.3	3.2
		Variable	120.1	115.9	48.1	343.6	1.4	169.3	124.7	923.2	3.9
3B	Los Angeles, CA	Constant	5.4	55.2	102.3	343.6	1.6	169.3	126.7	804.2	–
		Stepped	7.3	54.5	52.2	343.6	1.6	169.3	126.7	755.3	6.1
		Variable	7.3	56.8	46.5	343.6	1.6	169.3	126.6	751.7	6.5
3B	Las Vegas, NV	Constant	52.1	165.4	102.5	343.6	1.7	169.3	114.6	949.2	–
		Stepped	62.8	164.8	60.0	343.6	1.7	169.3	114.6	916.7	3.4
		Variable	62.7	163.2	50.9	343.6	1.7	169.3	114.6	905.9	4.6
3C	San Francisco, CA	Constant	45.2	16.8	102.1	343.6	1.6	169.3	118.8	797.4	–
		Stepped	56.0	15.6	45.9	343.6	1.6	169.3	118.9	750.9	5.8
		Variable	56.0	15.9	44.8	343.6	1.6	169.3	118.9	750.1	5.9
4A	Baltimore, MD	Constant	252.5	84.9	102.3	343.6	1.6	169.3	118.5	1,072.7	–
		Stepped	275.1	85.3	55.6	343.6	1.6	169.3	118.5	1,048.9	2.2
		Variable	275.5	87.2	49.2	343.6	1.6	169.3	118.3	1,044.7	2.6

Table B–3 Energy End Use Breakdown for Models With Refrigeration – SI Units (Continued)

Climate Zone	Location	Model	Gas ($MJ/m^2 \cdot yr$) Heating	Electricity ($MJ/m^2 \cdot yr$) Cooling	Fans	Interior Lights	Exterior Lights	Equipment	Refrigeration	Total ($MJ/m^2 \cdot yr$)	Savings (%)
4B	Albuquerque, NM	Constant	145.8	83.4	102.3	343.6	1.6	169.3	110.7	956.7	–
		Stepped	164.8	82.8	54.6	343.6	1.6	169.3	110.7	927.4	3.1
		Variable	165.1	82.9	47.3	343.6	1.6	169.3	110.6	920.4	3.8
4C	Seattle, WA	Constant	183.4	21.6	102.1	343.6	1.6	169.3	114.0	935.6	–
		Stepped	209.3	20.7	48.2	343.6	1.6	169.3	114.0	906.7	3.1
		Variable	209.5	21.2	46.4	343.6	1.6	169.3	114.0	905.6	3.2
5A	Chicago, IL	Constant	377.4	61.1	102.3	343.6	1.6	169.3	114.5	1,169.9	–
		Stepped	404.1	61.2	54.7	343.6	1.6	169.3	114.5	1,149.0	1.8
		Variable	404.4	62.9	49.2	343.6	1.6	169.3	114.4	1,145.4	2.1
5B	Boulder, CO	Constant	243.0	52.5	102.3	343.6	1.6	169.3	107.4	1,019.6	–
		Stepped	267.0	51.3	52.9	343.6	1.6	169.3	107.4	993.1	2.6
		Variable	266.6	51.2	48.3	343.6	1.6	169.3	107.3	988.0	3.1
6A	Minneapolis, MN	Constant	538.8	53.3	102.4	343.6	1.5	169.3	112.5	1,321.4	–
		Stepped	564.9	53.2	56.4	343.6	1.5	169.3	112.5	1,301.5	1.5
		Variable	565.1	54.8	51.3	343.6	1.5	169.3	112.4	1,298.0	1.8
6B	Helena, MT	Constant	411.7	33.4	102.3	343.6	1.5	169.3	104.7	1,166.5	–
		Stepped	441.1	32.4	52.8	343.6	1.5	169.3	104.7	1,145.4	1.8
		Variable	441.6	32.1	49.7	343.6	1.5	169.3	104.7	1,142.5	2.1
7A	Duluth, MN	Constant	673.8	20.3	102.3	343.6	1.6	169.3	106.3	1,417.3	–
		Stepped	706.3	19.8	54.6	343.6	1.6	169.3	106.3	1,401.5	1.1
		Variable	707.3	20.4	52.2	343.6	1.6	169.3	106.3	1,400.7	1.2
8A	Fairbanks, AK	Constant	1,158.9	9.6	102.6	343.6	1.5	169.3	101.5	1,887.0	–
		Stepped	1,188.3	8.8	59.9	343.6	1.5	169.3	101.5	1,873.0	0.7
		Variable	1,191.2	8.8	57.7	343.6	1.5	169.3	101.5	1,873.6	0.7

Table B–4 Energy End Use Breakdown for Models Without Refrigeration – SI Units

Climate Zone	Location	Model	Gas ($MJ/m^2 \cdot yr$)	Electricity ($MJ/m^2 \cdot yr$)					Total ($MJ/m^2 \cdot yr$)	Savings (%)
			Heating	Cooling	Fans	Interior Lights	Exterior Lights	Equipment		
1A	Miami, FL	Constant	0.3	278.8	85.1	341.6	1.8	146.1	853.7	–
		Stepped	0.4	285.9	58.7	341.6	1.8	146.1	834.4	2.3
		Variable	0.4	303.4	40.6	341.6	1.8	146.1	833.8	2.3
2A	Houston, TX	Constant	35.7	196.6	85.1	341.6	1.8	146.1	806.8	–
		Stepped	41.2	199.8	47.1	341.6	1.8	146.1	777.6	3.6
		Variable	41.7	212.2	34.3	341.6	1.8	146.1	777.6	3.6
2B	Phoenix, AZ	Constant	20.1	222.2	85.1	341.6	1.8	146.1	816.9	–
		Stepped	25.3	222.8	46.3	341.6	1.8	146.1	783.8	4.1
		Variable	25.3	225.0	35.1	341.6	1.8	146.1	774.8	5.2
3A	Atlanta, GA	Constant	110.2	111.5	85.1	341.6	1.7	146.1	796.2	–
		Stepped	123.6	114.4	39.1	341.6	1.7	146.1	766.5	3.7
		Variable	124.9	124.2	28.3	341.6	1.7	146.1	766.9	3.7
3B	Los Angeles, CA	Constant	6.3	52.3	85.1	341.6	1.9	146.1	633.3	–
		Stepped	8.4	52.3	31.3	341.6	1.9	146.1	581.7	8.1
		Variable	8.4	56.6	26.0	341.6	1.9	146.1	580.7	8.3
3B	Las Vegas, NV	Constant	51.7	170.3	85.1	341.6	2.0	146.1	796.7	–
		Stepped	63.0	170.1	41.2	341.6	2.0	146.1	764.0	4.1
		Variable	63.2	168.9	31.4	341.6	2.0	146.1	753.3	5.5
3C	San Francisco, CA	Constant	47.1	12.1	85.1	341.6	1.9	146.1	633.8	–
		Stepped	59.1	10.7	21.9	341.6	1.9	146.1	581.2	8.3
		Variable	59.1	10.9	21.2	341.6	1.9	146.1	580.8	8.4
4A	Baltimore, MD	Constant	261.1	84.3	85.1	341.6	1.9	146.1	920.0	–
		Stepped	283.4	85.8	38.7	341.6	1.9	146.1	897.4	2.5
		Variable	286.8	92.2	29.1	341.6	1.9	146.1	897.7	2.4

Table B–4 Energy End Use Breakdown for Models Without Refrigeration – SI Units (Continued)

Climate Zone	Location	Model	Gas ($MJ/m^2 \cdot yr$) Heating	Electricity ($MJ/m^2 \cdot yr$) Cooling	Fans	Interior Lights	Exterior Lights	Equipment	Total ($MJ/m^2 \cdot yr$)	Savings (%)
4B	Albuquerque, NM	Constant	147.9	82.8	85.1	341.6	1.9	146.1	805.3	–
		Stepped	167.0	82.8	37.1	341.6	1.9	146.1	776.5	3.6
		Variable	169.3	83.0	30.6	341.6	1.9	146.1	772.5	4.1
4C	Seattle, WA	Constant	190.7	17.5	85.1	341.6	1.9	146.1	782.8	–
		Stepped	219.7	16.6	25.8	341.6	1.9	146.1	751.6	4.0
		Variable	220.4	17.4	24.1	341.6	1.9	146.1	751.4	4.0
5A	Chicago, IL	Constant	391.7	59.5	85.1	341.6	1.9	146.1	1,025.9	–
		Stepped	414.7	60.4	40.0	341.6	1.9	146.1	1,004.7	2.1
		Variable	420.1	65.4	29.8	341.6	1.9	146.1	1,004.9	2.0
5B	Boulder, CO	Constant	248.1	50.4	85.1	341.6	1.9	146.1	873.2	–
		Stepped	270.0	49.4	36.1	341.6	1.9	146.1	845.1	3.2
		Variable	273.9	49.5	30.0	341.6	1.9	146.1	843.0	3.5
6A	Minneapolis, MN	Constant	559.2	51.6	85.2	341.6	1.8	146.1	1,185.5	–
		Stepped	579.8	52.3	43.2	341.6	1.8	146.1	1,164.8	1.7
		Variable	585.6	56.5	33.7	341.6	1.8	146.1	1,165.3	1.7
6B	Helena, MT	Constant	425.1	30.8	85.2	341.6	1.8	146.1	1,030.5	–
		Stepped	450.9	29.7	36.9	341.6	1.8	146.1	1,007.0	2.3
		Variable	456.1	29.6	30.9	341.6	1.8	146.1	1,006.1	2.4
7A	Duluth, MN	Constant	699.5	17.5	85.3	341.6	1.9	146.1	1,291.8	–
		Stepped	725.5	17.0	41.6	341.6	1.9	146.1	1,273.7	1.4
		Variable	732.2	18.5	34.2	341.6	1.9	146.1	1,274.4	1.3
8A	Fairbanks, AK	Constant	1,203.1	7.2	85.6	341.6	1.8	146.1	1,785.4	–
		Stepped	1,226.6	6.2	47.5	341.6	1.8	146.1	1,769.9	0.9
		Variable	1,232.0	6.3	42.2	341.6	1.8	146.1	1,770.0	0.9

B.2 Sizing

Table B–5 Cooling Coil Sizing Results by Zone for Models With Refrigeration – tons

Zone	Climate Zone															
	1A	2A	2B	3A	3BCA	3BNV	3C	4A	4B	4C	5A	5B	6A	6B	7	8
Grocery	43.3	43.3	43.3	43.3	43.3	43.3	43.3	43.3	43.3	43.3	43.3	43.3	43.3	43.3	43.3	43.3
Kitchen	4.8	4.8	4.8	4.8	4.8	4.8	4.8	4.8	4.8	4.8	4.8	4.8	4.8	4.8	4.8	4.8
Office	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0	12.0
Pharmacy	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2
Restroom	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4
Sales 1	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3
Sales 2	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3
Sales 3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3
Sales 4	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3
Stockroom	43.7	43.7	43.7	43.7	43.7	43.7	43.7	43.7	43.7	43.7	43.7	43.7	43.7	43.7	43.7	43.7
Vestibule	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4
Walk-in freezer	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2

Table B–6 Cooling Coil Sizing Results by Zone for Models With Refrigeration – ft^2/ton

Zone	Climate Zone															
	1A	2A	2B	3A	3BCA	3BNV	3C	4A	4B	4C	5A	5B	6A	6B	7	8
Grocery	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Kitchen	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Office	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Pharmacy	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Restroom	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Sales 1	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Sales 2	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Sales 3	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Sales 4	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Stockroom	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Vestibule	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Walk-in freezer	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375

Table B–7 Cooling Coil Sizing Results by Zone for Models With Refrigeration – kW

Zone	Climate Zone															
	1A	2A	2B	3A	3BCA	3BNV	3C	4A	4B	4C	5A	5B	6A	6B	7	8
Grocery	152	152	152	152	152	152	152	152	152	152	152	152	152	152	152	152
Kitchen	17	17	17	17	17	17	17	17	17	17	17	17	17	17	17	17
Office	42	42	42	42	42	42	42	42	42	42	42	42	42	42	42	42
Pharmacy	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
Restroom	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8
Sales 1	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205
Sales 2	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205
Sales 3	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205
Sales 4	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205
Stockroom	154	154	154	154	154	154	154	154	154	154	154	154	154	154	154	154
Vestibule	19	19	19	19	19	19	19	19	19	19	19	19	19	19	19	19
Walk-in freezer	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18

Table B–8 Cooling Coil Sizing Results by Zone for Models Without Refrigeration – tons

Zone	Climate Zone															
	1A	2A	2B	3A	3BCA	3BNV	3C	4A	4B	4C	5A	5B	6A	6B	7	8
Office	22.0	22.0	22.0	22.0	22.0	22.0	22.0	22.0	22.0	22.0	22.0	22.0	22.0	22.0	22.0	22.0
Restroom	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4
Sales 1	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3
Sales 2	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3
Sales 3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3
Sales 4	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3	58.3
Stockroom	35.1	35.1	35.1	35.1	35.1	35.1	35.1	35.1	35.1	35.1	35.1	35.1	35.1	35.1	35.1	35.1
Vestibule	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4	5.4

Table B–9 Cooling Coil Sizing Results by Zone for Models Without Refrigeration – ft^2/ton

Zone	Climate Zone															
	1A	2A	2B	3A	3BCA	3BNV	3C	4A	4B	4C	5A	5B	6A	6B	7	8
Office	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Restroom	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Sales 1	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Sales 2	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Sales 3	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Sales 4	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Stockroom	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Vestibule	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375

Table B–10 Cooling Coil Sizing Results by Zone for Models Without Refrigeration – kW

Zone	Climate Zone															
	1A	2A	2B	3A	3BCA	3BNV	3C	4A	4B	4C	5A	5B	6A	6B	7	8
Office	77	77	77	77	77	77	77	77	77	77	77	77	77	77	77	77
Restroom	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8
Sales 1	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205
Sales 2	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205
Sales 3	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205
Sales 4	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205	205
Stockroom	123	123	123	123	123	123	123	123	123	123	123	123	123	123	123	123
Vestibule	19	19	19	19	19	19	19	19	19	19	19	19	19	19	19	19

Table B–11 Heating Coil Sizing Results by Zone for Models With Refrigeration – kBtu/h

Zone	Climate Zone															
	1A	2A	2B	3A	3BCA	3BNV	3C	4A	4B	4C	5A	5B	6A	6B	7	8
Grocery	804	839	790	828	806	776	821	869	712	838	889	740	901	809	893	973
Kitchen	94	101	94	102	94	94	97	108	88	102	114	94	117	105	117	131
Office	219	226	214	222	219	209	223	232	191	225	235	196	237	212	234	252
Pharmacy	101	110	102	110	102	101	106	117	96	111	123	102	127	114	127	143
Restroom	47	51	47	51	47	47	49	54	44	51	57	47	58	53	58	66
Sales 1	1,138	1,233	1,140	1,237	1,147	1,137	1,183	1,313	1,070	1,240	1,380	1,143	1,420	1,280	1,417	1,595
Sales 2	1,138	1,233	1,140	1,237	1,147	1,137	1,183	1,313	1,070	1,240	1,380	1,143	1,420	1,280	1,417	1,595
Sales 3	1,138	1,233	1,140	1,237	1,147	1,137	1,183	1,313	1,070	1,240	1,380	1,143	1,420	1,280	1,417	1,595
Sales 4	1,138	1,233	1,140	1,237	1,147	1,137	1,183	1,313	1,070	1,240	1,380	1,143	1,420	1,280	1,417	1,595
Stockroom	813	850	800	840	815	785	831	881	722	849	902	751	915	822	907	991
Vestibule	100	104	98	102	100	96	102	107	88	104	109	91	111	99	110	119
Walk-in freezer	101	110	102	110	102	101	105	117	95	111	123	102	127	114	126	142

Table B–12 Heating Coil Sizing Results by Zone for Models With Refrigeration – Btu/h·ft^2

Zone	Climate Zone															
	1A	2A	2B	3A	3BCA	3BNV	3C	4A	4B	4C	5A	5B	6A	6B	7	8
Grocery	49.5	51.6	48.6	51.0	49.6	47.7	50.5	53.5	43.8	51.5	54.7	45.5	55.4	49.8	54.9	59.9
Kitchen	52.0	56.4	52.1	56.6	52.4	52.0	54.1	60.0	48.9	56.7	63.1	52.3	64.9	58.5	64.8	72.9
Office	48.7	50.2	47.6	49.3	48.8	46.5	49.5	51.5	42.3	50.0	52.2	43.5	52.6	47.2	52.0	56.0
Pharmacy	52.1	56.4	52.2	56.7	52.5	52.0	54.1	60.1	49.0	56.8	63.2	52.4	65.1	58.6	65.0	73.1
Restroom	52.0	56.4	52.1	56.6	52.4	52.0	54.1	60.0	48.9	56.7	63.1	52.3	64.9	58.5	64.8	72.9
Sales 1	52.0	56.4	52.1	56.6	52.4	52.0	54.1	60.0	48.9	56.7	63.1	52.3	64.9	58.5	64.8	72.9
Sales 2	52.0	56.4	52.1	56.6	52.4	52.0	54.1	60.0	48.9	56.7	63.1	52.3	64.9	58.5	64.8	72.9
Sales 3	52.0	56.4	52.1	56.6	52.4	52.0	54.1	60.0	48.9	56.7	63.1	52.3	64.9	58.5	64.8	72.9
Sales 4	52.0	56.4	52.1	56.6	52.4	52.0	54.1	60.0	48.9	56.7	63.1	52.3	64.9	58.5	64.8	72.9
Stockroom	49.6	51.8	48.8	51.2	49.7	47.9	50.7	53.7	44.0	51.7	55.0	45.8	55.8	50.1	55.3	60.4
Vestibule	49.3	51.2	48.3	50.5	49.4	47.4	50.2	52.9	43.4	51.1	54.0	45.0	54.7	49.1	54.1	58.8
Walk-in freezer	52.0	56.4	52.1	56.6	52.4	52.0	54.1	60.0	48.9	56.7	63.1	52.3	64.9	58.5	64.8	72.9

Table B–13 Heating Coil Sizing Results by Zone for Models With Refrigeration – kW

Zone	Climate Zone															
	1A	2A	2B	3A	3BCA	3BNV	3C	4A	4B	4C	5A	5B	6A	6B	7	8
Grocery	236	246	232	243	236	227	241	255	209	245	260	217	264	237	262	285
Kitchen	27	30	27	30	28	27	29	32	26	30	33	28	34	31	34	38
Office	64	66	63	65	64	61	65	68	56	66	69	57	69	62	69	74
Pharmacy	30	32	30	32	30	30	31	34	28	32	36	30	37	34	37	42
Restroom	14	15	14	15	14	14	14	16	13	15	17	14	17	15	17	19
Sales 1	333	361	334	363	336	333	347	385	314	363	405	335	416	375	415	468
Sales 2	333	361	334	363	336	333	347	385	314	363	405	335	416	375	415	468
Sales 3	333	361	334	363	336	333	347	385	314	363	405	335	416	375	415	468
Sales 4	333	361	334	363	336	333	347	385	314	363	405	335	416	375	415	468
Stockroom	238	249	234	246	239	230	244	258	212	249	264	220	268	241	266	290
Vestibule	29	30	29	30	29	28	30	31	26	30	32	27	32	29	32	35
Walk-in freezer	30	32	30	32	30	30	31	34	28	32	36	30	37	33	37	42

Table B–14 Heating Coil Sizing Results by Zone for Models Without Refrigeration – kBtu/h

Zone	Climate Zone															
	1A	2A	2B	3A	3BCA	3BNV	3C	4A	4B	4C	5A	5B	6A	6B	7	8
Office	338	350	331	345	338	324	344	361	296	349	368	306	372	334	368	399
Restroom	40	44	40	44	40	40	42	47	38	44	50	41	52	47	52	59
Sales 1	967	1063	976	1073	977	979	1013	1143	929	1072	1214	1003	1254	1132	1255	1427
Sales 2	967	1063	976	1073	977	979	1013	1143	929	1072	1214	1003	1254	1132	1255	1427
Sales 3	967	1063	976	1073	977	979	1013	1143	929	1072	1214	1003	1254	1132	1255	1427
Sales 4	967	1063	976	1073	977	979	1013	1143	929	1072	1214	1003	1254	1132	1255	1427
Stockroom	549	579	543	575	552	535	564	605	495	579	623	518	635	571	630	693
Vestibule	84	88	83	87	84	81	86	91	75	88	94	78	95	86	95	104

Table B–15 Heating Coil Sizing Results by Zone for Models Without Refrigeration – Btu/h·ft^2

Zone	Climate Zone															
	1A	2A	2B	3A	3BCA	3BNV	3C	4A	4B	4C	5A	5B	6A	6B	7	8
Office	40.9	42.4	40.1	41.8	41.0	39.3	41.7	43.8	35.9	42.3	44.5	37.1	45.1	40.4	44.6	48.3
Restroom	44.2	48.6	44.6	49.1	44.7	44.8	46.3	52.3	42.5	49.0	55.5	45.9	57.3	51.7	57.4	65.3
Sales 1	44.2	48.6	44.6	49.1	44.7	44.8	46.3	52.3	42.5	49.0	55.5	45.9	57.3	51.7	57.4	65.3
Sales 2	44.2	48.6	44.6	49.1	44.7	44.8	46.3	52.3	42.5	49.0	55.5	45.9	57.3	51.7	57.4	65.3
Sales 3	44.2	48.6	44.6	49.1	44.7	44.8	46.3	52.3	42.5	49.0	55.5	45.9	57.3	51.7	57.4	65.3
Sales 4	44.2	48.6	44.6	49.1	44.7	44.8	46.3	52.3	42.5	49.0	55.5	45.9	57.3	51.7	57.4	65.3
Stockroom	41.8	44.0	41.3	43.7	42.0	40.7	42.9	46.0	37.6	44.1	47.4	39.4	48.3	43.4	47.9	52.7
Vestibule	41.5	43.5	40.8	43.0	41.6	40.2	42.4	45.2	37.0	43.4	46.4	38.6	47.1	42.3	46.7	51.1

Table B–16 Heating Coil Sizing Results by Zone for Models Without Refrigeration – kW

Zone	Climate Zone															
	1A	2A	2B	3A	3BCA	3BNV	3C	4A	4B	4C	5A	5B	6A	6B	7	8
Office	99	103	97	101	99	95	101	106	87	102	108	90	109	98	108	117
Restroom	12	13	12	13	12	12	12	14	11	13	15	12	15	14	15	17
Sales 1	283	312	286	314	286	287	297	335	272	314	356	294	368	332	368	418
Sales 2	283	312	286	314	286	287	297	335	272	314	356	294	368	332	368	418
Sales 3	283	312	286	314	286	287	297	335	272	314	356	294	368	332	368	418
Sales 4	283	312	286	314	286	287	297	335	272	314	356	294	368	332	368	418
Stockroom	161	170	159	168	162	157	165	177	145	170	183	152	186	167	185	203
Vestibule	25	26	24	26	25	24	25	27	22	26	28	23	28	25	28	30

B.3 Fan Energy Savings (SI Units)

Table B–17 Normalized Annual Fan Energy Consumption for Models With Refrigeration

Climate Zones	Cities	Constant-Speed Fan	Stepped-Speed Fan		Variable-Speed Fan	
		Fan EUI (MJ/m^2)	Fan EUI (MJ/m^2)	Savings (%)	Fan EUI (MJ/m^2)	Savings (%)
1A	Miami, FL	102.3	72.8	28.9	57.0	44.3
2A	Houston, TX	102.1	64.0	37.3	53.5	47.7
2B	Phoenix, AZ	102.2	63.9	37.5	53.1	48.0
3A	Atlanta, GA	102.0	56.8	44.4	47.9	53.1
3B	Los Angeles, CA	102.0	52.0	49.0	46.3	54.6
3B	Las Vegas, NV	102.1	59.8	41.4	50.7	50.4
3C	San Francisco, CA	101.8	45.8	55.1	44.6	56.2
4A	Baltimore, MD	101.9	55.4	45.7	49.0	51.9
4B	Albuquerque, NM	102.0	54.4	46.6	47.2	53.7
4C	Seattle, WA	101.8	48.0	52.9	46.3	54.5
5A	Chicago, IL	101.9	54.5	46.6	49.0	51.9
5B	Boulder, CO	101.9	52.7	48.3	48.2	52.7
6A	Minneapolis, MN	102.0	56.2	44.9	51.1	49.9
6B	Helena, MT	101.9	52.6	48.4	49.5	51.4
7	Duluth, MN	101.9	54.4	46.7	52.1	48.9
8	Fairbanks, AK	102.2	59.7	41.6	57.5	43.8

Table B–18 Normalized Annual Fan Energy Consumption for Models Without Refrigeration

Climate Zones	Cities	Constant-Speed Fan	Stepped-Speed Fan		Variable-Speed Fan	
		Fan EUI (MJ/m^2)	Fan EUI (MJ/m^2)	Savings (%)	Fan EUI (MJ/m^2)	Savings (%)
1A	Miami, FL	84.8	58.5	31.0	40.5	52.2
2A	Houston, TX	84.8	46.9	44.6	34.2	59.7
2B	Phoenix, AZ	84.8	46.1	45.6	34.9	58.8
3A	Atlanta, GA	84.8	38.9	54.1	28.2	66.7
3B	Los Angeles, CA	84.8	31.2	63.2	25.9	69.4
3B	Las Vegas, NV	84.8	41.1	51.5	31.3	63.1
3C	San Francisco, CA	84.8	21.8	74.3	21.2	75.0
4A	Baltimore, MD	84.8	38.5	54.6	29.0	65.9
4B	Albuquerque, NM	84.8	37.0	56.3	30.5	64.1
4C	Seattle, WA	84.8	25.7	69.7	24.0	71.7
5A	Chicago, IL	84.8	39.9	53.0	29.7	65.0
5B	Boulder, CO	84.8	36.0	57.6	29.9	64.7
6A	Minneapolis, MN	84.9	43.1	49.3	33.6	60.4
6B	Helena, MT	84.9	36.8	56.7	30.8	63.7
7	Duluth, MN	85.0	41.5	51.2	34.1	59.9
8	Fairbanks, AK	85.3	47.3	44.5	42.1	50.7

B.4 Electricity Consumption Savings (SI Units)

Table B–19 Normalized Annual Whole-Building Electricity Consumption for Models With Refrigeration

Climate Zones	Cities	Constant-Speed Fan	Stepped-Speed Fan		Variable-Speed Fan	
		Use (kWh/m^2)	Use (kWh/m^2)	Savings (kWh/m^2)	Use (kWh/m^2)	Savings (kWh/m^2)
1A	Miami, FL	285.4	278.2	7.1	275.3	10.1
2A	Houston, TX	261.4	251.2	10.2	249.2	12.2
2B	Phoenix, AZ	265.0	254.2	10.7	251.0	14.0
3A	Atlanta, GA	236.3	224.1	12.3	222.3	14.0
3B	Los Angeles, CA	221.2	207.1	14.1	206.1	15.1
3B	Las Vegas, NV	248.4	236.4	11.9	233.5	14.9
3C	San Francisco, CA	208.3	192.4	15.9	192.2	16.1
4A	Baltimore, MD	227.1	214.2	12.8	213.0	14.1
4B	Albuquerque, NM	224.5	211.1	13.4	209.1	15.4
4C	Seattle, WA	208.3	193.1	15.2	192.7	15.5
5A	Chicago, IL	219.4	206.2	13.2	205.2	14.2
5B	Boulder, CO	215.0	201.0	14.0	199.7	15.3
6A	Minneapolis, MN	216.7	203.9	12.7	202.9	13.7
6B	Helena, MT	209.0	195.0	14.0	194.1	14.9
7	Duluth, MN	205.8	192.5	13.4	192.0	13.9
8	Fairbanks, AK	201.6	189.6	12.0	189.0	12.6

Table B–20 Normalized Annual Whole-Building Electricity Consumption for Models Without Refrigeration

Climate Zones	Cities	Constant-Speed Fan	Stepped-Speed Fan		Variable-Speed Fan	
		Use (kWh/m^2)	Use (kWh/m^2)	Savings (kWh/m^2)	Use (kWh/m^2)	Savings (kWh/m^2)
1A	Miami, FL	236.3	230.9	5.4	230.8	5.5
2A	Houston, TX	213.5	203.9	9.6	203.7	9.8
2B	Phoenix, AZ	220.6	210.0	10.6	207.5	13.1
3A	Atlanta, GA	189.9	178.0	12.0	177.7	12.2
3B	Los Angeles, CA	173.6	158.7	14.9	158.4	15.2
3B	Las Vegas, NV	206.3	194.1	12.2	191.0	15.2
3C	San Francisco, CA	162.4	144.5	17.9	144.4	18.0
4A	Baltimore, MD	182.4	170.0	12.5	169.1	13.3
4B	Albuquerque, NM	182.0	168.7	13.3	167.0	15.0
4C	Seattle, WA	163.9	147.3	16.6	147.0	16.9
5A	Chicago, IL	175.6	163.4	12.2	161.9	13.7
5B	Boulder, CO	173.1	159.2	13.8	157.6	15.5
6A	Minneapolis, MN	173.4	162.0	11.4	160.5	12.9
6B	Helena, MT	167.6	154.0	13.7	152.3	15.3
7	Duluth, MN	164.0	151.8	12.2	150.1	13.9
8	Fairbanks, AK	161.2	150.4	10.8	148.9	12.3

B.5 Natural Gas Consumption Savings (SI Units)

Table B–21 Normalized Annual Whole-Building Natural Gas Consumption* for Models With Refrigeration

Climate Zones	Cities	Constant-Speed Fan	Stepped-Speed Fan		Variable-Speed Fan	
		Use (MJ/m^2)	Use (MJ/m^2)	Savings (MJ/m^2)	Use (MJ/m^2)	Savings (MJ/m^2)
1A	Miami, FL	0.2	0.3	0.0	0.3	0.0
2A	Houston, TX	33.9	39.1	–5.3	39.2	–5.3
2B	Phoenix, AZ	20.0	24.5	–4.6	24.6	–4.6
3A	Atlanta, GA	106.7	120.0	–13.3	120.1	–13.4
3B	Los Angeles, CA	5.4	7.3	–2.0	7.3	–2.0
3B	Las Vegas, NV	52.1	62.8	–10.7	62.7	–10.7
3C	San Francisco, CA	45.2	56.0	–10.8	56.0	–10.8
4A	Baltimore, MD	252.5	275.1	–22.5	275.5	–22.9
4B	Albuquerque, NM	145.8	164.8	–19.0	165.1	–19.3
4C	Seattle, WA	183.4	209.3	–26.0	209.5	–26.1
5A	Chicago, IL	377.4	404.1	–26.7	404.4	–26.9
5B	Boulder, CO	243.0	267.0	–24.1	266.6	–23.7
6A	Minneapolis, MN	538.8	564.9	–26.1	565.1	–26.3
6B	Helena, MT	411.7	441.1	–29.5	441.6	–29.9
7	Duluth, MN	673.8	706.3	–32.5	707.3	–33.5
8	Fairbanks, AK	1,158.9	1,188.3	–29.4	1,191.2	–32.3

*Negative values indicate that natural gas use increased.

Table B–22 Normalized Annual Whole-Building Natural Gas Consumption* for Models Without Refrigeration

Climate Zones	Cities	Constant-Speed Fan	Stepped-Speed Fan		Variable-Speed Fan	
		Use (MJ/m^2)	Use (MJ/m^2)	Savings (MJ/m^2)	Use (MJ/m^2)	Savings (MJ/m^2)
1A	Miami, FL	0.3	0.4	0.0	0.4	0.0
2A	Houston, TX	35.7	41.2	–5.5	41.7	–6.0
2B	Phoenix, AZ	20.1	25.3	–5.1	25.3	–5.2
3A	Atlanta, GA	110.2	123.6	–13.4	124.9	–14.8
3B	Los Angeles, CA	6.3	8.4	–2.1	8.4	–2.1
3B	Las Vegas, NV	51.7	63.0	–11.3	63.2	–11.5
3C	San Francisco, CA	47.1	59.1	–12.0	59.1	–12.0
4A	Baltimore, MD	261.1	283.4	–22.3	286.8	–25.8
4B	Albuquerque, NM	147.9	167.0	–19.1	169.3	–21.4
4C	Seattle, WA	190.7	219.7	–29.0	220.4	–29.7
5A	Chicago, IL	391.7	414.7	–22.9	420.1	–28.4
5B	Boulder, CO	248.1	270.0	–21.8	273.9	–25.8
6A	Minneapolis, MN	559.2	579.8	–20.7	585.6	–26.5
6B	Helena, MT	425.1	450.9	–25.8	456.1	–31.0
7	Duluth, MN	699.5	725.5	–26.0	732.2	–32.7
8	Fairbanks, AK	1,203.1	1,226.6	–23.5	1,232.0	–28.9

*Negative values indicate that natural gas use increased.

Appendix C How to Calculate Preliminary Utility Cost Savings

Retail building managers and engineers can use this appendix to calculate preliminary utility cost savings associated with stepped- and variable-speed RTU fan motor retrofits. These high-level results are approximate, but can be used to justify site-specific assessments of retrofit feasibility. They may also help to present a compelling business case for fan motor upgrades.

Use Table C–1 and the following steps to estimate the annual whole-building utility cost savings for your buildings.

Table C–1 Annual Whole Building Energy Savings (%) for All Models

Climate Zone	Location	Models With Refrigeration		Models Without Refrigeration	
		Stepped-Speed	Variable-Speed	Stepped-Speed	Variable-Speed
1A	Miami, FL	2.5	3.5	2.3	2.3
2A	Houston, TX	3.2	4.0	3.6	3.6
2B	Phoenix, AZ	3.5	4.7	4.1	5.2
3A	Atlanta, GA	3.2	3.9	3.7	3.7
3B	Los Angeles, CA	6.1	6.5	8.1	8.3
3B	Las Vegas, NV	3.4	4.6	4.1	5.5
3C	San Francisco, CA	5.8	5.9	8.3	8.4
4A	Baltimore, MD	2.2	2.6	2.5	2.4
4B	Albuquerque, NM	3.1	3.8	3.6	4.1
4C	Seattle, WA	3.1	3.2	4.0	4.0
5A	Chicago, IL	1.8	2.1	2.1	2.0
5B	Boulder, CO	2.6	3.1	3.2	3.5
6A	Minneapolis, MN	1.5	1.8	1.7	1.7
6B	Helena, MT	1.8	2.1	2.3	2.4
7	Duluth, MN	1.1	1.2	1.4	1.3
8	Fairbanks, AK	0.7	0.7	0.9	0.9

Step 1: Select the climate zone that best matches your building location.

Step 2: If your building has significant refrigeration loads, choose values from the columns labeled "Models With Refrigeration." Otherwise, choose values from the columns labeled "Models Without Refrigeration."

Step 3: Select the fan motor functionality for which you are interested in estimating savings. Then identify the whole-building annual energy savings value for that case from Table C–1.

Step 4: Use Table C–2 to calculate the utility cost savings for your building.

Table C–2 High-Level Calculation Methodology

Annual Building Energy Costs ($/year)		Whole-Building Annual Energy Savings (%)		Annual Energy Cost Savings ($/year)
	X		=	